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HIGH TEMPERATURE HYDRAULIC SYSTEM
ACTUATOR SEALS FOR USE IN ADVANCED
SUPERSONIC AIRCRAFT

by

J. Lee

prepared for

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

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SECOND SEMIANNUAL REPORT

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ABSTRACT

This report covers the second six month period of a program to investigate seal materials and to design seals for high temperature hydraulic actuator application. Work has progressed in the design and fabrication of the seal test rigs, materials evaluation, seal design and selection of candidate seal designs. The test rig for evaluating one-inch rod seals has been fabricated and checked out at temperatures ranging from room temperature to 600°F. Evaluation of seal materials is essentially complete and selection of candidate materials for further evaluation has been accomplished. Selection of candidate seal designs for further development has also been accomplished.

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Fairchild Hiller
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SUMMARY

This report describes activities completed during the second six-month period of NASA Contract NAS 3-7264, ending 31 March 1966. The object of this program is to develop hydraulic actuator seals intended to function reliably for 3000 hours in the temperature range of -40 to +600°F. In order to achieve this objective, the work being performed provides for the investigation of advanced materials and the development and test of seal concepts.

As of this reporting period, progress was made in the following areas:

- Design, fabrication and checkout the seal cycling rig
- Fluid compatibility, mechanical properties, and sliding wear test of seal materials
- Selection of candidate seal materials
- Design, test, and selection of candidate seal designs
- Evaluation of candidate seal designs

Checkout of the one-inch seal cycling rig at room temperature, 400°F, 500°F, and 600°F has been accomplished. Operation of the rig with Polymer SP V-seals in the test actuator was satisfactory. Work on the three-inch seal test rig has been initiated. Evaluation of seal materials for fluid compatibility, mechanical properties and sliding wear characteristics is essentially complete. Five materials were selected and approved by NASA for further investigation. These were: silver-copper alloy, nickel Foametal impregnated with $\text{CaF}_2 + \text{BaF}_2$, Vascojet 1000,

cobalt molybdenum alloy, and Polymer SP.

Additional efforts were devoted to developing concepts for the second-stage seal. Tests conducted on a metallic lip seal fabricated of Vascojet 1000 indicated zero leakage operation up to 400°F. Low friction is one of the attractive characteristics of this configuration.

A rating system was developed to aid in the evaluation of the numerous seal concepts. Based on this rating system, four basic seal designs were selected from approximately 17 candidates for further development. The four basic designs provide for the following five seal-material combinations:

- 1) V-seal with Polymer SP
- 2) Lip seal with Vascojet 1000
- 3) Lip seal with cobalt molybdenum
- 4) Wedge seal with nickel Foametal
- 5) Reed seal with silver alloy and Vascojet 1000 or silver alloy with cobalt molybdenum

Detail design and testing have been initiated of the V-seal and two lip seal configurations.

INTRODUCTION

The concept of sustained supersonic flight has become a reality within the past several years and the trend for the future is toward the development of even higher speed aircraft. As operating conditions become more severe with succeeding families of vehicles, design margins will decrease substantially. Not the least to be affected by these considerations are hydraulic system components, particularly the dynamic seals. Elastomeric seals are now meeting the requirements of current aircraft operating in the temperature range of -65°F to +275°F. However, the temperature extremes anticipated for future air vehicles will impose significantly greater demands on these materials and will limit their use. For example, recent research programs on high temperature seals, conducted by Republic and summarized in ASD-TDR-63-573 and ML-TDR-64-266, indicate that the maximum operating

temperature permissible with present day elastomers is 400°F.

The objective of this present program is to investigate advanced materials and seal concepts for potential use in fluid power systems of future supersonic aircraft. This investigation is therefore directed to dynamic rod seals intended to function efficiently for 3000 hours in the temperature range of -40°F to +600°F, and operating pressures to 4000 psi.

Emphasis is placed on integrating material properties and seal design to obtain the optimum seal-material combination. The specific tasks to be accomplished are:

- Task I - Preparation of existing test facilities and design and fabrication of seal test actuators and fixtures.
- Task II - Selection, procurement and evaluation of candidate seal materials.
- Task III - Design of seals for the one and three-inch rod sizes.
- Task IV - Low pressure testing of one and three-inch rod seals at temperatures of 400°F, 500°F and 600°F.
- Task V - Long-term testing of the most promising seal-material combinations in the one and three-inch rod sizes.
- Task VI - Development and evaluation of a single-stage high-pressure rod seal in the one-inch rod size.

Detailed discussion of the progress made in above tasks is presented in the following sections.

TASK I - FACILITIES AND EQUIPMENT

A. GENERAL

Activities within this task were centered on the design and fabrication of the one-inch and three-inch seal test rigs. All seal test actuators for these rigs have been received. The one-inch seal test rig has been completed and checked out at room temperature to 600°F. Work on the three-inch test rig is approximately 40 percent complete. Calibration of strain gauges for measuring seal friction has been initiated.

B. SEAL TEST ACTUATORS

Three each of the one-inch and three-inch seal test actuators have been fabricated. These will be used in both the low pressure (Task IV) and high pressure (Task V) test phases. During the fabrication of these actuators, some difficulties were encountered in obtaining the proper surface finish on the piston rods because of the limitation of the vendor's equipment. Consequently the piston rods were finished by another source.

A typical test actuator is shown in Figures 1 and 2. The actuator consists of a double-ended cylinder with removable seal glands on each end. The actuator housing is fabricated of 17-4PH corrosion-resistant steel. Piston rods are fabricated of Type 440C stainless steel and plated with hard chromium. The seal cavity is designed to accommodate various seal configurations. Leakage ports are located so that leakage from the static and dynamic portion of the rod seal can be measured separately. A barrier seal is provided in the cylinder to simulate the pressure conditions in a typical actuator. For example, during piston rod extension, chamber No. 1 senses high pressures while chamber No. 2 is vented to return. During piston rod retraction, the pressure conditions would be reversed. The advantage of this design is that the piston head is not required, thus eliminating the need for heavy structural members to externally load the actuator.

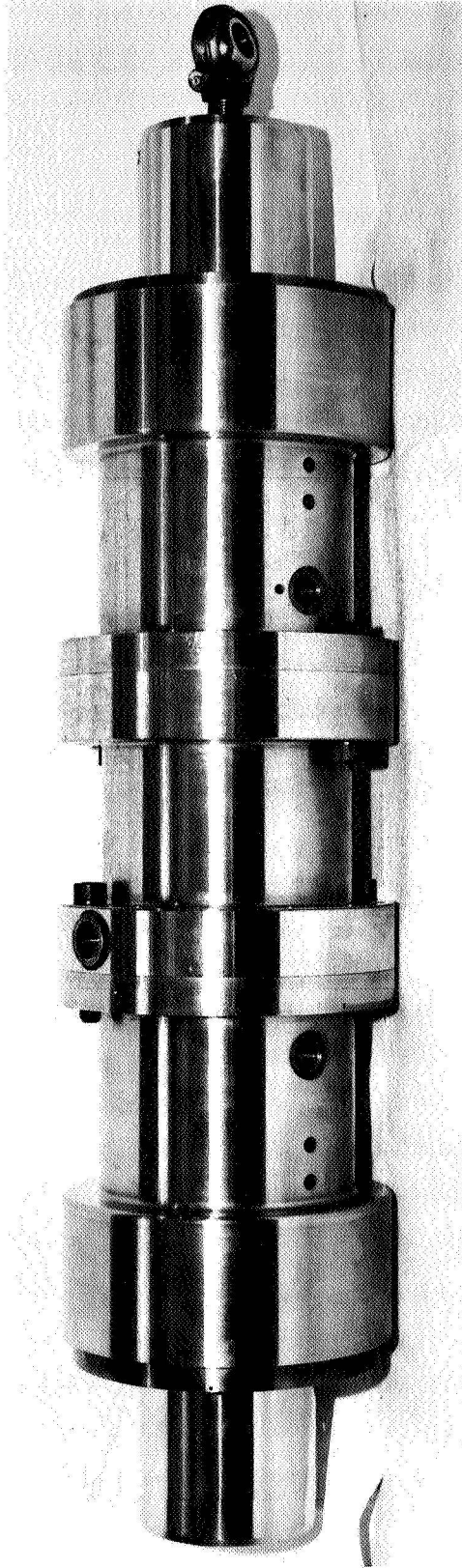


Figure 1. Test Actuator

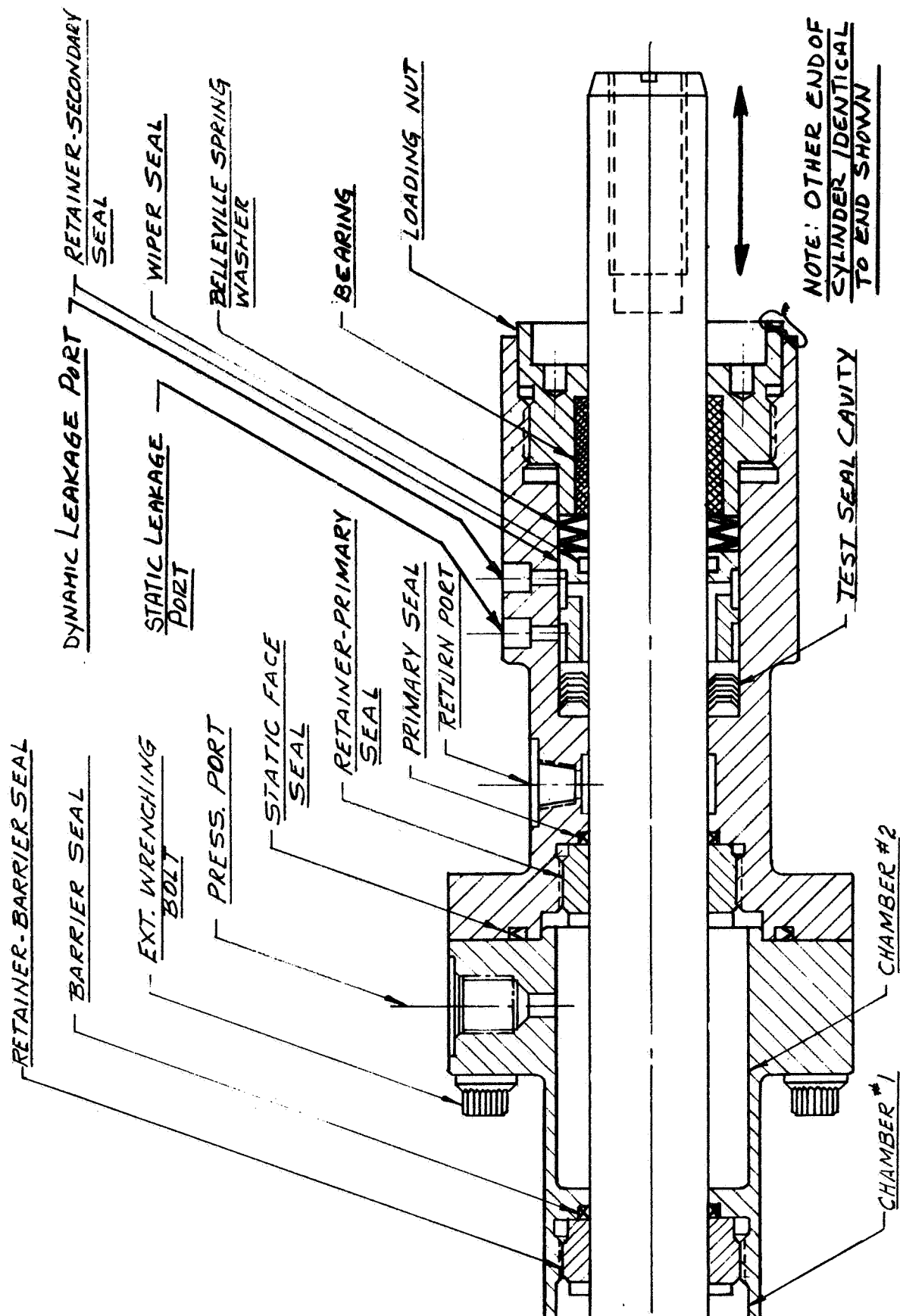


Figure 2. One-Inch Rod Seal Test Actuator

C. SEAL TEST RIG

Fabrication of the one-inch cycling rig for low pressure testing has been completed. As shown in Figure 3, the rig consists of a seal test actuator and driving actuator. The test actuator is trunion mounted in the oven and driven through a bell crank. The driving actuator located outside of the oven is a mechanical input servo controlled type. The servo valve input link is connected to an eccentric cam which is driven by a variable speed motor. Thus, the length of the piston rod stroke and cycling rate can be varied by adjusting the cam radius and motor speed, respectively.

A schematic of the hydraulic system for the test rig is shown in Figure 4. The system consists of a power circuit and test circuit. The power circuit supplies pressure from a hydraulic mule to the driving actuator. The test circuit is comprised mainly of a booster pump, which is used for filling, and an accumulator for maintaining fluid pressure during testing. Leakage lines from the test actuator are brought outside of the oven to facilitate constant monitoring during testing. Thermocouples are strategically placed to measure ambient conditions, actuator skin temperature and seal temperature. Thermocouples for monitoring seal temperatures are placed inside the actuator adjacent to the test seal.

Operational checkout of the test rig has been accomplished at room temperature, 400°F, 500°F, and 600°F. The test rig was operated for a total of 23 hours. During the operation the seal test actuator, which was assembled with Polymer SP V-seals, was cycled at rates of 20 cpm (± 2 -inch stroke) and 300 cpm ($\pm 1/8$ -inch stroke). Pressure in the actuator was 100 psi. Operation of the rig and test actuator was satisfactory. However, it was found that a circulating fan was needed in the oven chamber to provide better heat distribution. The strain gauge for monitoring seal friction was calibrated at room temperature and is now installed in the actuator for high temperature checkout.

Work on the three-inch cycling rig is now in progress.

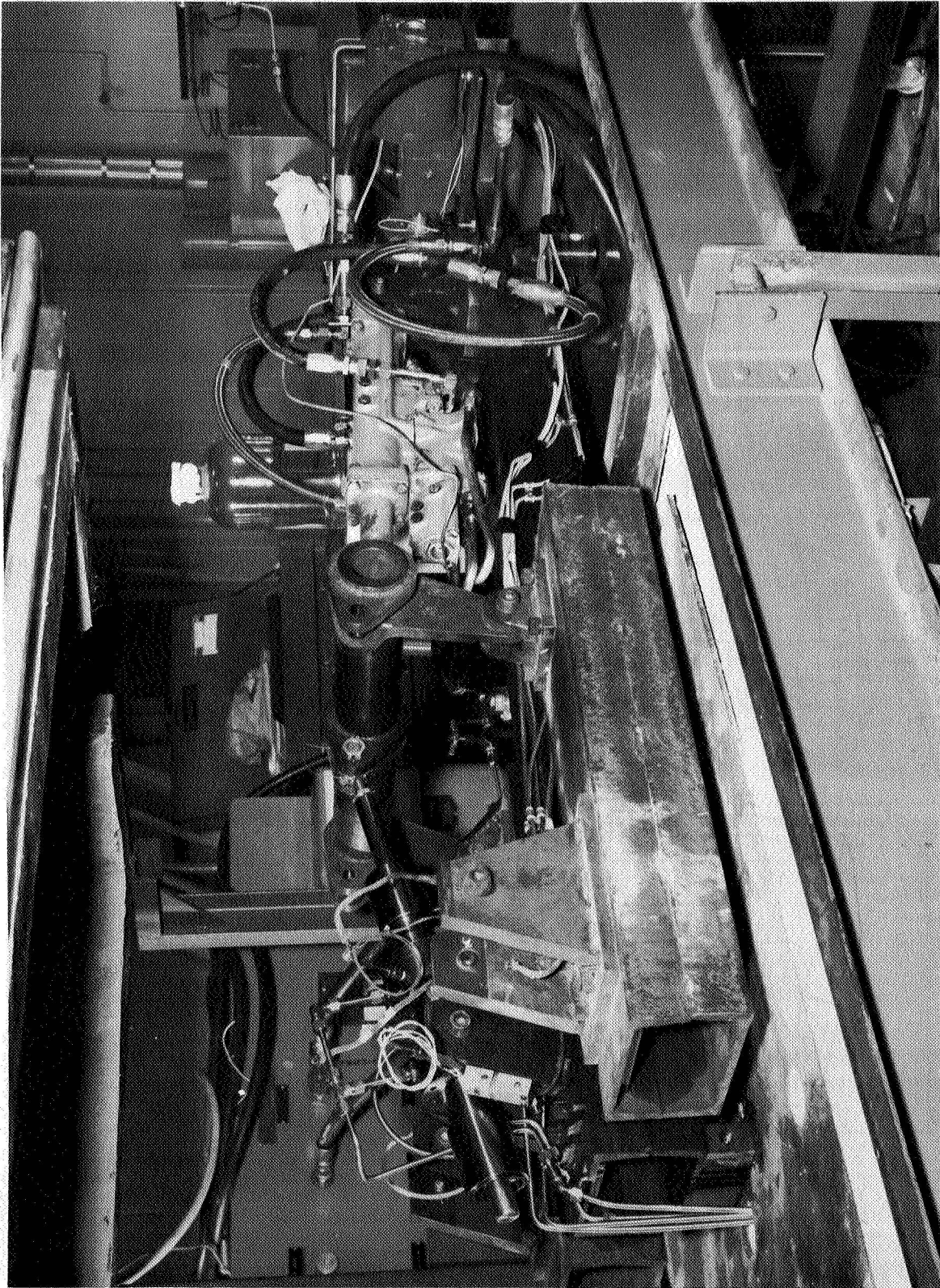


Figure 3. One-inch Cycling Rig

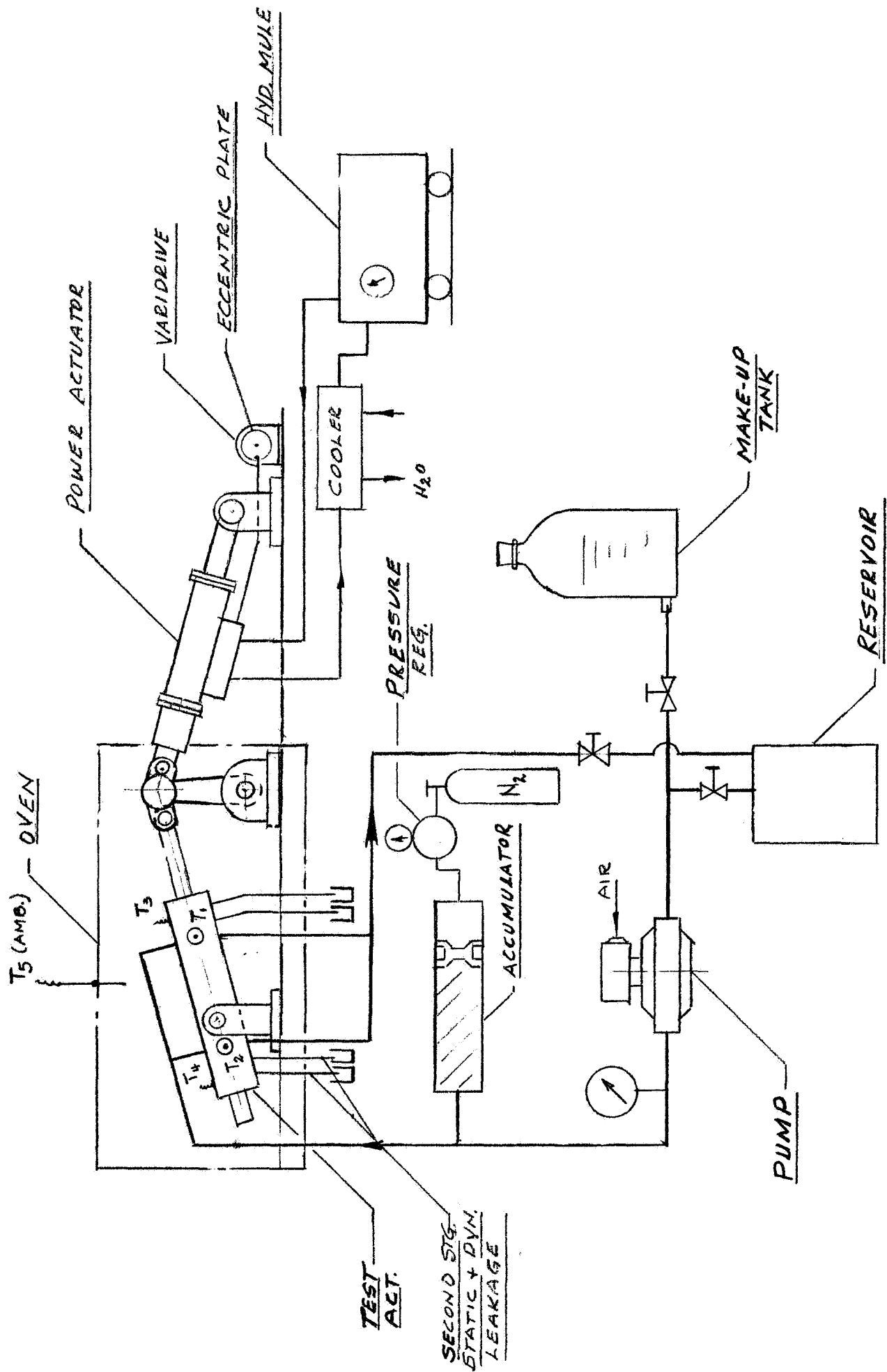


Figure 4. Hydraulic System Schematic

D. STRAIN GAUGE CALIBRATION

The strain gauge for measuring seal friction, a temperature compensated type, is manufactured by Microdot Inc. As shown in Figure 5, the gauge is mounted on the shank of the rod end bearing. The rod end bearing is in turn attached to the piston rod of the seal test actuator and the driving bell crank.

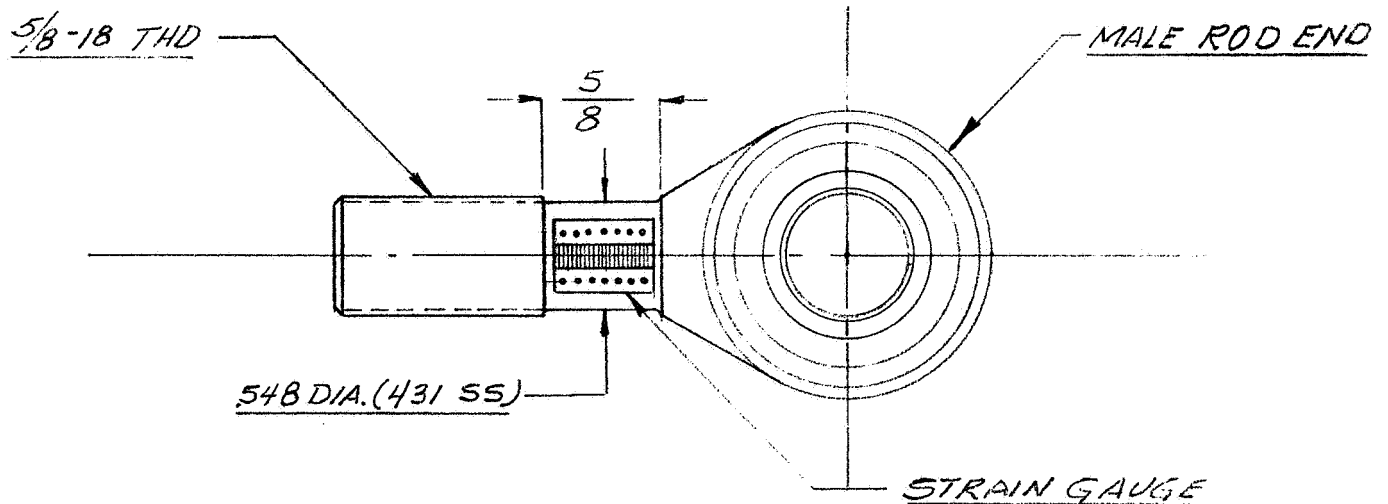


Figure 5. Strain Gauge - Rod End Installation

Calibration of the strain gauge was accomplished on a lathe as shown in Figure 6. The rod end was held firmly in the tail stock while the load cell was held in the chuck. A steel bolt connected the rod end to the load cell.

Figure 7 shows the strain in micro-inches versus load, obtained both in tension and compression. This curve was obtained by applying loads in increments of 100 pounds to the rod end, and measuring the strain in micro-inches corresponding to the various loads. It was noted that the strain versus load curve in compression is much steeper than the corresponding curve for tensile loads. This additional strain in compression was caused by bending loads which were induced in the rod end. These bending loads may be balanced out by the addition of a second strain gage located diametrically opposite to the existing one on the shank of the rod end bearing. A mathematical check of the tensile strain at 200 pound load is shown on the following page.

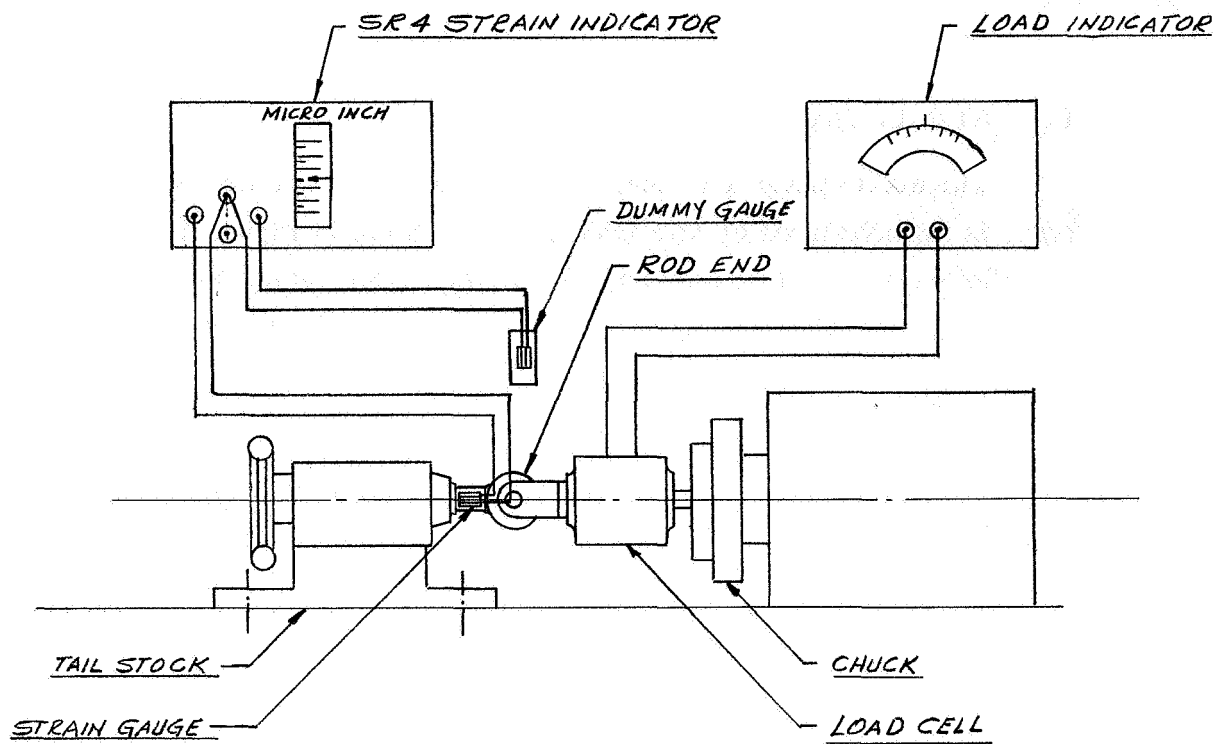


Figure 6. Strain Gauge Calibration Set-Up

$$P = 200 \text{ lb.}$$

$$E = 29,000,000$$

$$A = .235$$

$$L = \text{effective gauge length} = .50 \text{ inch}$$

$$\frac{P}{A} = S = 850 \text{ lb.}$$

$$E = \frac{PL}{A\epsilon} = \frac{SL}{\epsilon}$$

$$\epsilon = \frac{850 \times .50 \text{ in.}}{29,000,000}$$

$$\epsilon = .0000146 \text{ inch}$$

This value approximates very closely the strain at 200 pounds for the tension part of the load versus deflection curve.

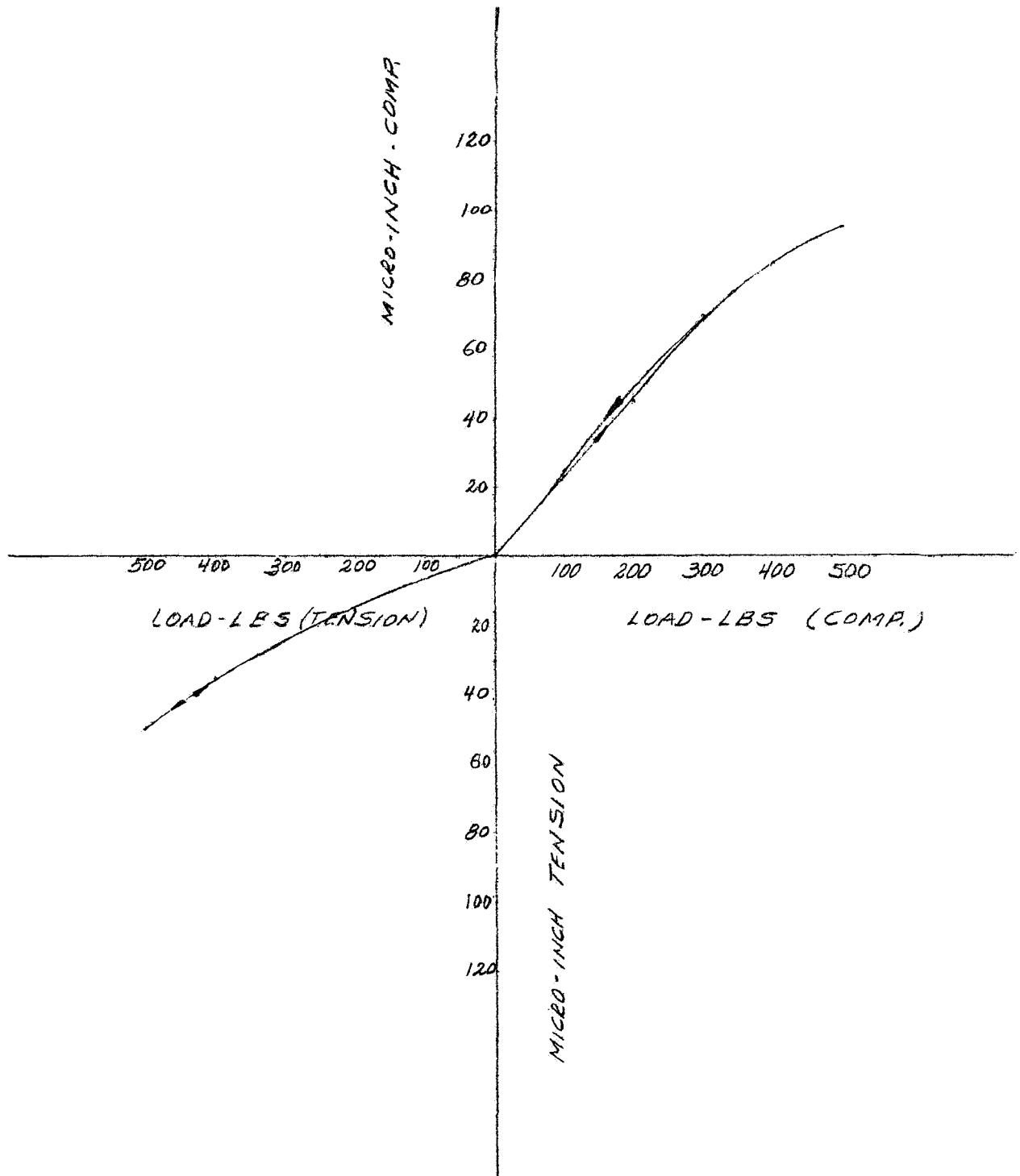


Figure 7. Load versus Deflection - .548-inch Diameter Rod End

TASK II - MATERIALS EVALUATION

A. GENERAL

Fluid compatibility, sliding wear tests, and mechanical properties tests were accomplished on essentially all of the candidate seal materials.

The candidate seal materials are listed below.

- 1) Unfilled Polymer SP. Alternates: (a) 15% graphite filled, (b) 30% bronze filled, (c) 20% copper filled.
- 2) Polymet. Alternate: Westinghouse 75% silver, 20% polyimide, 5% tungsten diselenide composite.
- 3) Silver impregnated stainless steel fiber composite. Alternate: silver impregnated nickel fiber composite.
- 4) Silver alloy (72% silver, 28% copper). Alternate: silver alloy (60% silver, 40% copper).
- 5) Nickel Foametal (60% density, impregnated with an eutectic mixture of calcium fluoride and barium fluoride).
- 6) Westinghouse 70% silver, 30% tungsten diselenide composite.
- 7) Cobalt alloy (75% cobalt, 25% molybdenum). This material supplied by NASA.
- 8) Titanium alloy (84% titanium, 16% tin). This material supplied by NASA.
- 9) Metco flame-plated molybdenum (burnished with molybdenum disulfide).
- 10) Tool steel (Vascojet 1000).

Fluids tested in combination with the materials cited above were the same fluids under investigation in Republic's Hydraulic Fluids Evaluation Program (Ref. 1). These fluids are shown on the following page.

- 1) F-50 Silicone
- 2) MCS-3101 Halogenated Polyaryl
- 3) MCS-293 Modified Polyphenyl Ether
- 4) PR-143-AB Fluorocarbon
- 5) MLO-60-294 Super Refined Mineral Oil

B. FLUID-MATERIAL COMPATIBILITY

Fluid-material compatibility tests were sequentially conducted at 600°F, 400°F, and 500°F with the candidate fluids and seal materials. The test apparatus, as previously described in NASA CR-54496, is shown in Figures 8 and 9. To prevent intermixing of fluids between manifolds, the apparatus was modified by replacing the check valves with positive shut-off manual valves.

As shown in Figure 10, the test specimen, which is approximately 7/8-inch in diameter, is kept in face contact with a polished, hard chromium-plated stainless steel button by means of a screw and spring assembly. This arrangement provides a constant contact pressure between the test specimen and button at elevated temperatures. Hardness readings were taken on the bulk material prior to machining the specimens. The specimens were weighed prior to testing. Each specimen was inserted in a capsule as shown in Figure 11. The capsules were fabricated from one-inch diameter stainless steel tubing. Approximately 25 ml of fluid were used in each capsule.

1. Fluid-material Compatibility at 600°F

For the 600°F run the following materials were evaluated:

- a) Westinghouse composite (70% silver, 30% tungsten diselenide)
- b) Silver alloy (72% silver, 28% copper)
- c) Westinghouse composite (70% silver, 25% polyimide, and 5% tungsten diselenide) - alternate material
- d) Silver-stainless steel (Type 430) composite
- e) Silver-nickel composite - alternate material
- f) Nickel Foametal impregnated with calcium fluoride and barium fluoride
- g) Vascojet 1000 (H-11 tool steel)

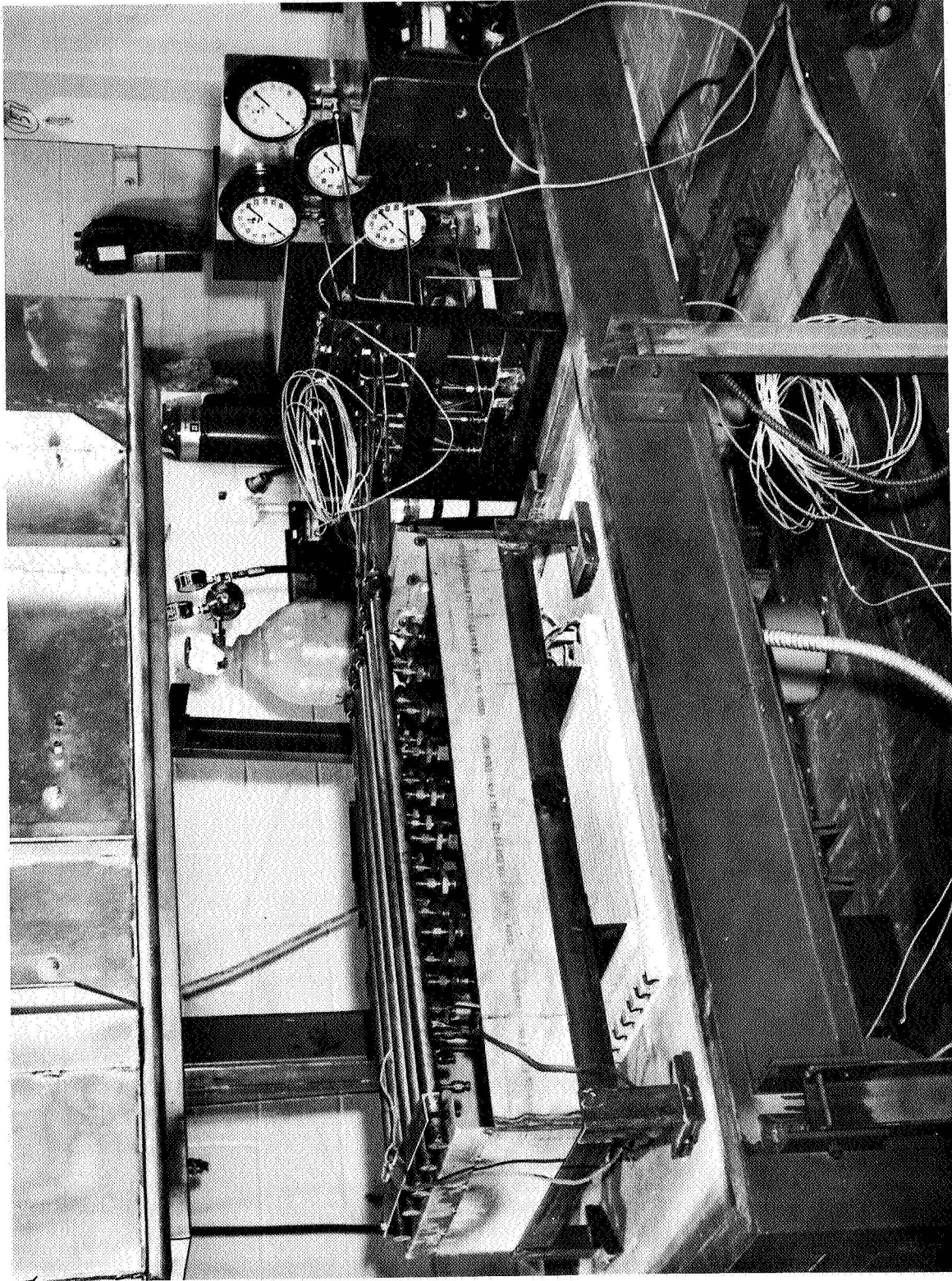


Figure 8. Test Set-up for Compatibility Tests

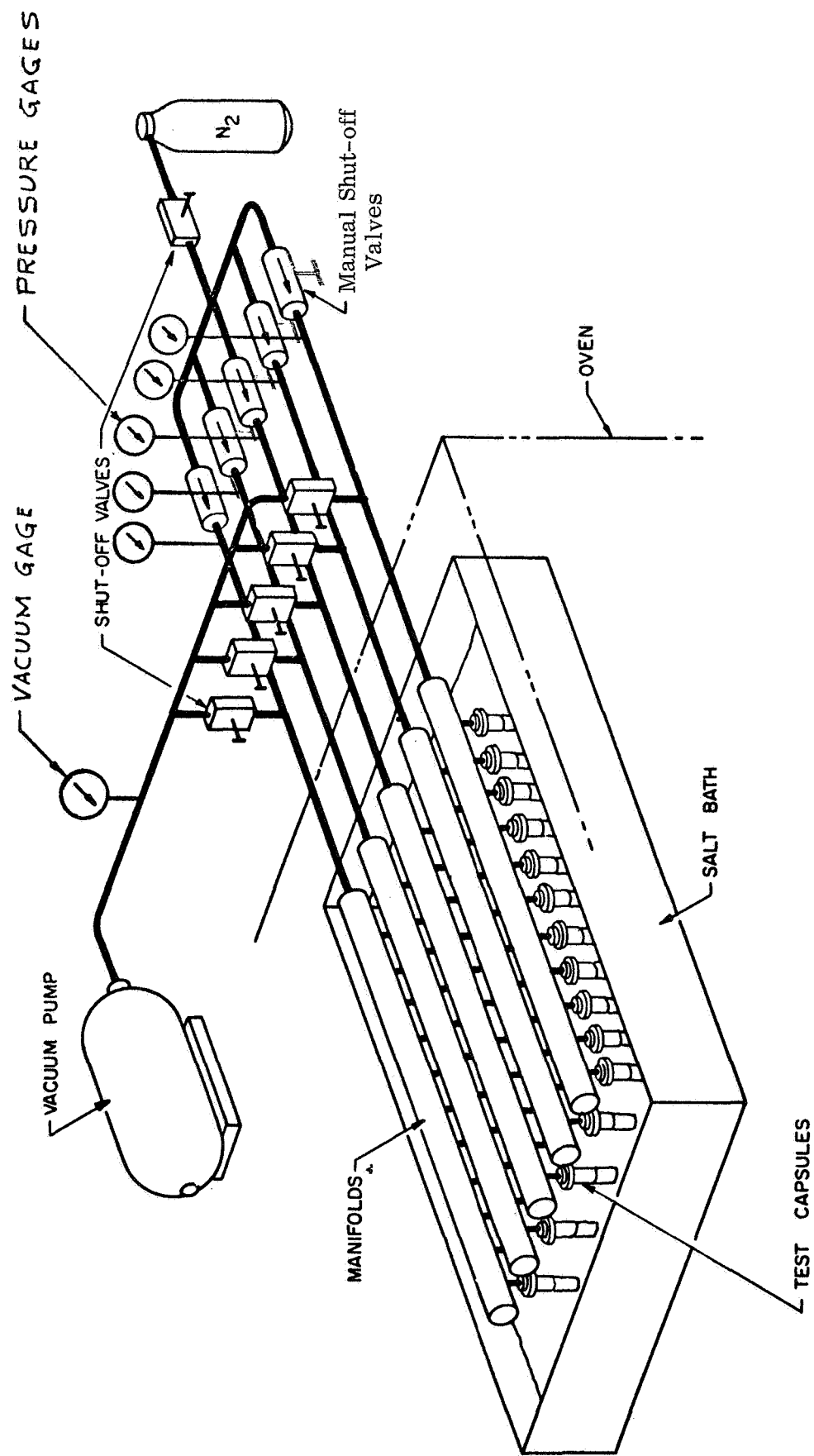


Figure 9. Schematic - Test Setup for Compatibility Tests

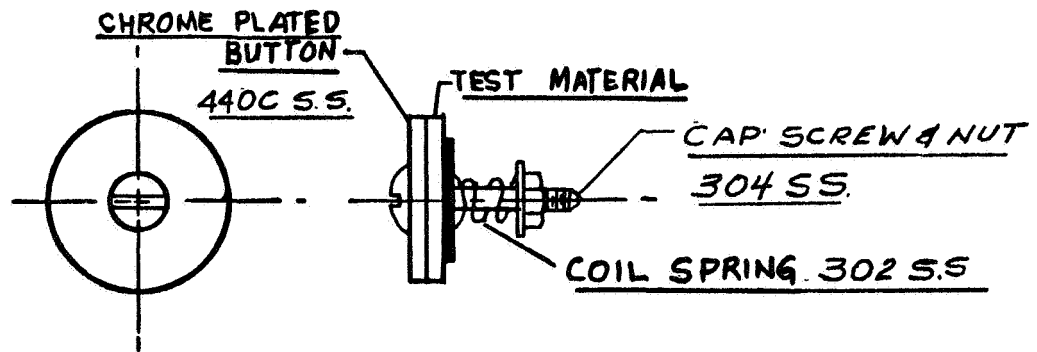


Figure 10. Test Specimen Assembly

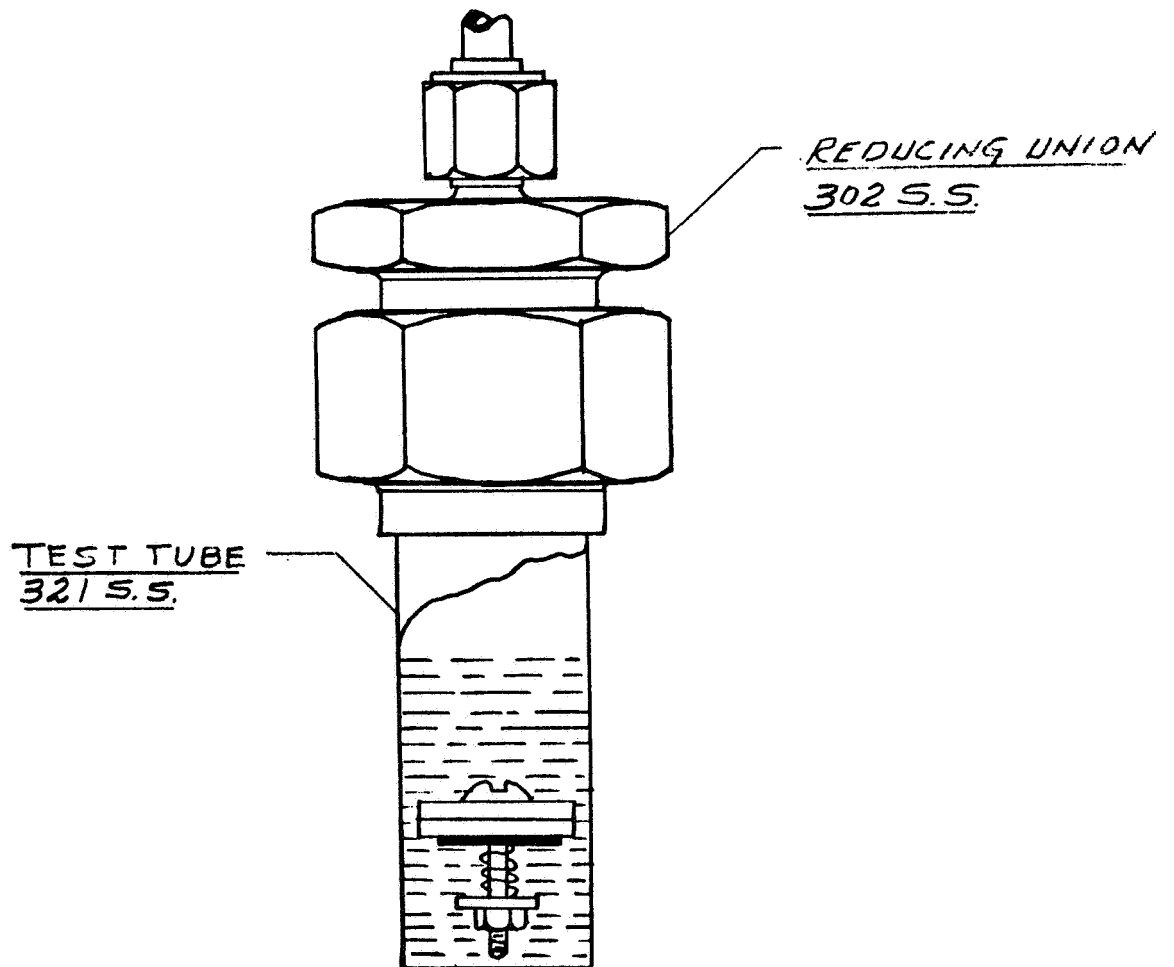


Figure 11. Test Capsule

- h) Titanium tin alloy
- i) Cobalt molybdenum alloy
- j) Metco flame-plated molybdenum (titanium base material)

The above materials were tested with F-50, MCS-293, PR 143AB, and MLO 60-294 fluids. Polymer SP and Polymet, which are prime candidate materials, were not included in the test with the foregoing fluids since previous testing (Ref. 2), indicated that they were compatible. This provided an opportunity to obtain data on two alternate materials. The MCS-3101, fluid which exhibited poor thermal stability in previous testing at 600°F (Ref. 2), was virtually eliminated from further consideration as a 600°F fluid. However, a limited number of MCS-3101 fluid samples were included in the current tests to verify results obtained in the previous test. These samples were evaluated in four separate test tubes apart from the main test apparatus. One tube contained the MCS-3101 fluid as the control; the other three tubes contained Polymer SP, Polymet, and the silver-stainless steel (Type 430) composite, respectively.

All the fluids completed the 150 hour test at 600°F. General condition of the fluids and material specimens are shown in Figures 12 through 15. Changes in the viscosity and acidic condition of the fluid samples, and hardness changes of the materials specimens are summarized in Tables 1 to 4. In general, the results indicate that fairly good compatibility exist between the candidate seal materials and PR-143AB, MCS-293 and MLO-60-294 fluids. The exceptions were as follows:

- a) Westinghouse composite (70% silver + 30% tungsten diselenide) was not compatible with the F-50 silicone. The fluid crystallized in the tube.
- b) Silver-stainless steel (Type 430) composite exhibited slight corrosion with the F-50 silicone.
- c) Vascojet 1000 exhibited slight corrosion when exposed to air after being in contact with F-50 silicone.

The F-50 silicone fluid appears to be unstable at 600°F. A considerable increase in viscosity and acidity was exhibited by this fluid. The control fluid, in particular, showed a viscosity of 997 centistokes at 100°F, as compared to the


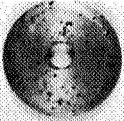



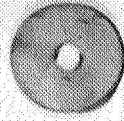
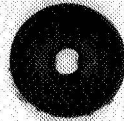
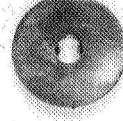
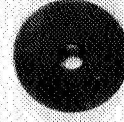
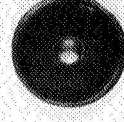
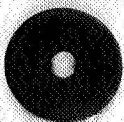
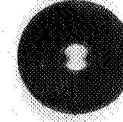
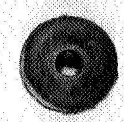
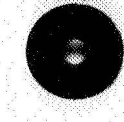
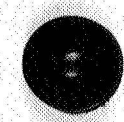
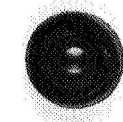
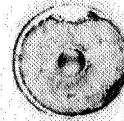
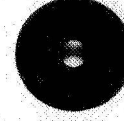
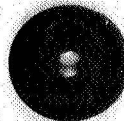
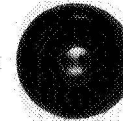
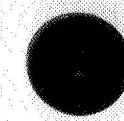
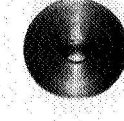
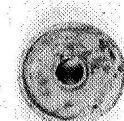
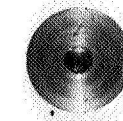
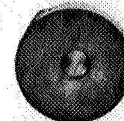
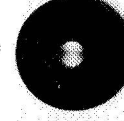

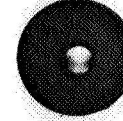


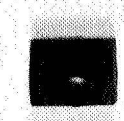
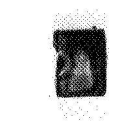
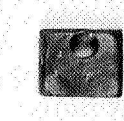
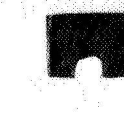
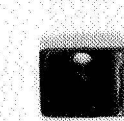
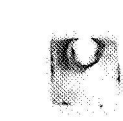
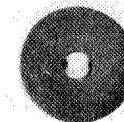
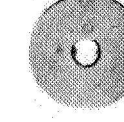
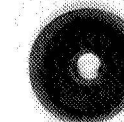
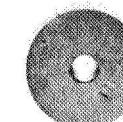
	F-50 Silicone	PR-143AB	MCS-293	MLO-60-294
70% Ag. 25% polyimide, 5% tungsten diselenide				
Silver alloy 72% Ag + 28% Cu.				
70% Ag 30% tungsten diselenide				
Silver - S.S. Composite				
Silver- Nickel Composite				
Nickel Foametal Impregnated w/ CaF_2 + BaF_2				
Vascojet 1000				
Titanium - tin alloy (10% tin)				
Cobalt- molybdenum alloy				
Flame-plated Molybdenum Coating				

Figure 12. Test No. 2 - Material Specimens After Test at 600°F

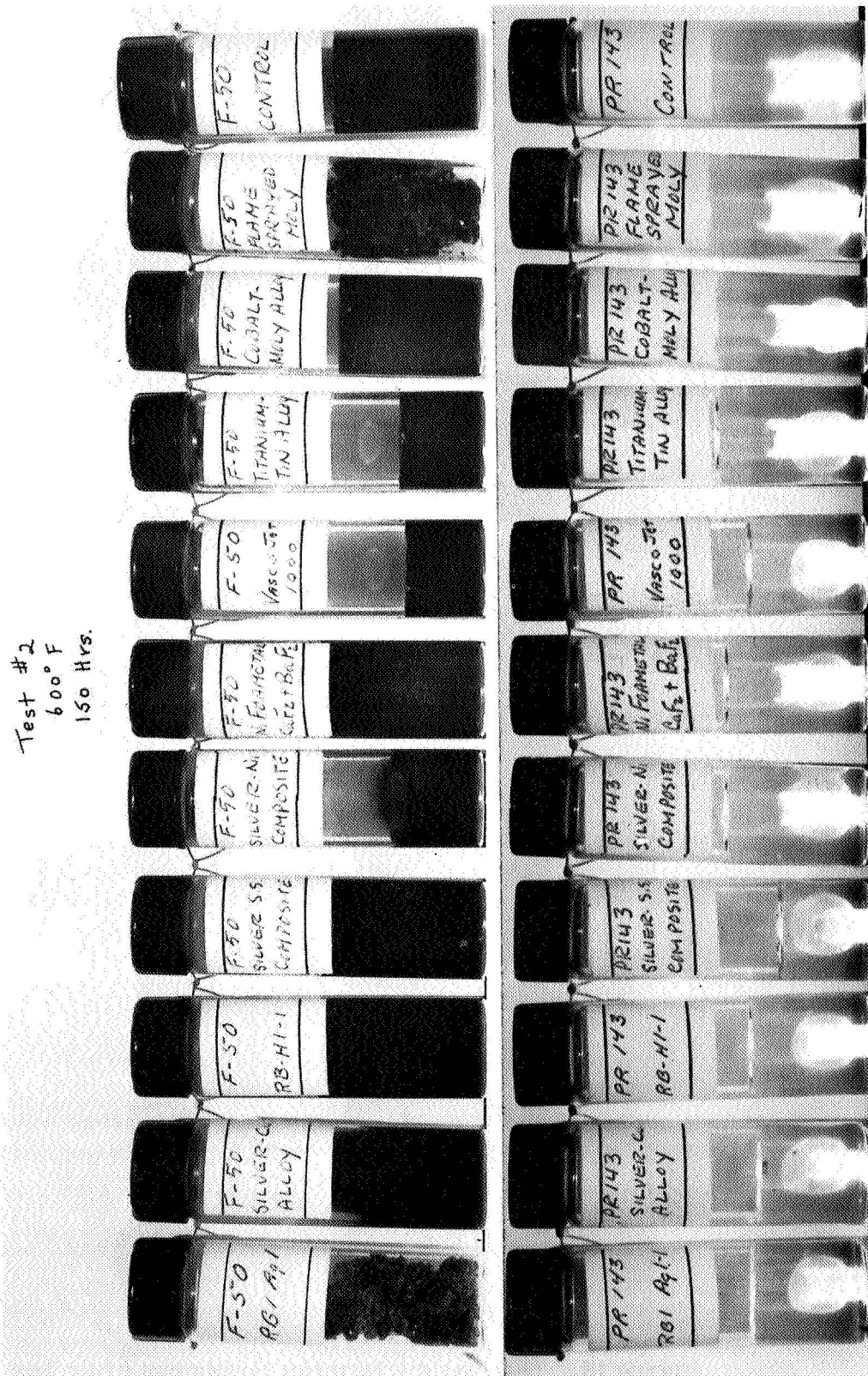


Figure 13. Test No. 2 - F-50 Silicone and PR-143AB After Test at 600°F

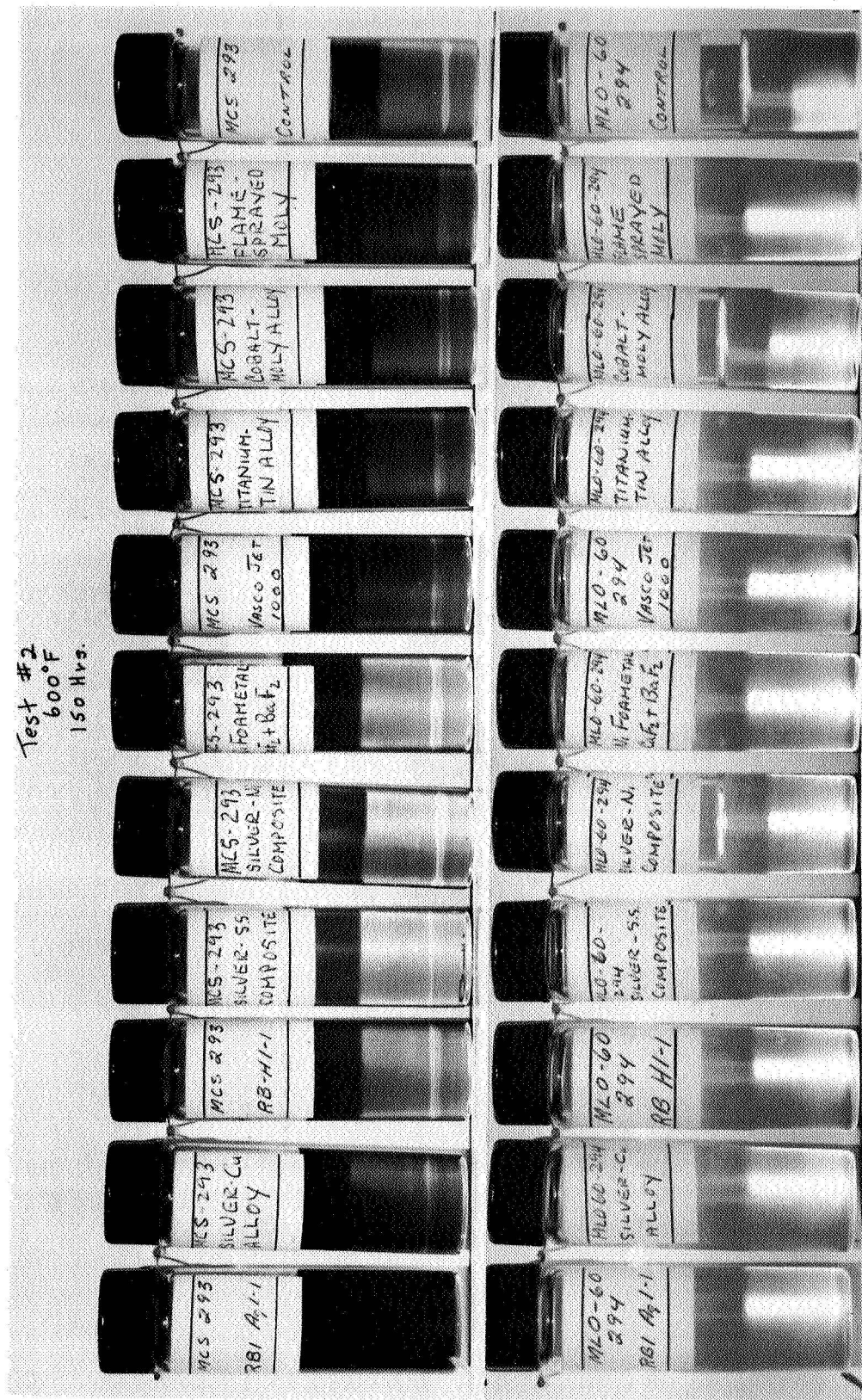


Figure 14. Test No. 2 - MCS-293 and MLO-60-294 After Test at 600°F



Figure 15. Test No. 2 - MCS-3101 After Test at 600°F

TABLE 1
FLUID - MATERIAL COMPATIBILITY TEST NO. 2 - 150 HOURS AT 600°F
F-50 SILICONE FLUID
25 ml Per Test Specimen

Specimen	70% Ag. 25% Polyimide +5% Tungsten Diselenide	72% Ag. + 28% Copper	70% Ag. 30% Tungsten Diselenide	Silver - S.S. Composite (430 S.S.)	Silver-Nickel Composite	Nickel Foametal Impregnated w/Ca F2 + Ba F2	Vascojet 1000	Titanium- Tin Alloy	Cobalt- Molybdenum Alloy	Plasma Flame-Plated Molybdenum	Control
Weight Change (Grams)	-.0287	+.0020	+.0166	+.0185	+.1566	+.0149	-.0004	+.0010	+.0046	+.0016	
Hardness Change	H85 to H66	F89 to F91	H94 to H93.5	H90 to H96.5	H83 to H98	F87 to F92	RC19 to RC16	RC36 to RC345	RC5 to RC2	RC39 to RC37	
Appearance of Specimen	Dull grey finish on both sides	Copper color on both sides	Dull grey finish on both sides. Crystallized fluid deposit on side oppo- site of mating surface.	Brown deposit on mating side, discolored on opposite side.	Brown deposit on mating side, white coating on opposite side.	Thin dark deposit on both sides.	No change on mating side, thin coating on opposite side.	Slightly dis- colored on both sides	No change on mating side, thin dark coating on opposite side.	No change on mating side, slightly dis- colored on opposite side.	
Appearance of Mating Surface	Thin green deposit on chrome side, dark deposit on unplated side.		Green deposit on chrome side, deposit of crystallized fluid on un- plated side.	Thin green deposit on plated side. Thin dark coating on opposite side.	No change on chrome side, thin dark coating on opposite side	Slight discoloration on chrome side, thin dark coating on opposite side.	No change on mating side. Thin dark coating on opposite side.	No change on mating side. Thin dark coating on opposite side.	No change on mating side. Thin dark coating on opposite side.	Plated side discolored, thin dark coating on opposite side.	
Appearance of Fluid	Dark brown	Dark brown	Fluid crystallized	Dark brown	Dark brown	Dark brown	Dark brown	Dark brown	Dark brown	Dark brown	Dark brown
Viscosity @ 100°F (48.18)	102.00	133.50	*	98.00	277.50	113.80	113.50	102.70	92.60	107.00	997.0
Viscosity @ 210°F (15.99)	33.77	45.58	*	32.40	102.16	35.27	39.29	29.03	29.33	33.61	249.70
Acid No. mg KOH/g (0.03)	4.23	19.0	-	8.66	1.16	20.2	8.88	8.1	7.31	11.8	2.04

* Fluid crystallized

TABLE 2
FLUID - MATERIAL COMPATIBILITY TEST NO. 2 - 150 HOURS AT 600°F
PR-143AB
25 ml Per Test Specimen

Specimen	70% Ag. 25% Polyimide +5% Tungsten Diselenide	72% Ag. + 28% Copper	70% Ag. 30% Tungsten Diselenide	Silver - S.S. Composite (430 S.S.)	Silver-Nickel Composite	Nickel Foametal Impregnated w/ Ca F2 + Ba F2	Vascojet 1000	Titanium- Tin Alloy	Cobalt- Molybdenum Alloy	Plasma Flame-plated Molybdenum	Control
Weight Change	-.0184	+.0026	+.0503	+.0090	+.0070	+.0067	+.0014	+.0039	+.0030	+.0004	
Hardness Change	H85 to H71	F89 to F89.5	H94 to H99	H90 to H92.5	H83 to H82	F87 to F91.5	RC19 to RC18.5	RC36 to RC37	RC5 to RC6.5	RC39 to RC36.5	
Appearance of Specimen	Dulled finish on both sides	Copper color on both sides.	Thin dark coating on mating side,	Thin dark coating on mating side.	Thin dark coating on mating side,	Dulled finish on both sides.	Both sides slightly discolored.	Mating side discolored, slight de- posit on opposite side.	No change on both sides.	No change on mating side, thin dark deposit on opposite side.	
Appearance of Mating Surface	No change on plated side, unplated side discolored.	Both sides discolored.	Both sides discolored.	Plated side discolored, unplated side darkened.	Plated side slightly pitted, un- plated side discolored.	No change on plated side, unplated side discolored.	Both sides discolored.	Both sides slightly discolored.	Both sides discolored.	Both sides discolored.	
Appearance of Fluid	No change	No change	No change	No change	No change	No change	No change	No change	No change	No change	
Viscosity @ 100°F C.S. (60.13)	68.70	76.50	68.40	67.70	67.45	68.50	69.60	68.25	68.60	67.80	68.3
Viscosity @ 210°F C.S. (71.94)	8.63	9.44	8.67	8.48	8.48	8.84	8.79	8.71	8.72	8.88	8.80
Acid No. mg KOH/g (.01)	.12	.02	.03	.06	.06	.03	.06	.09	.03	.06	.09

TABLE 3
FLUID - MATERIAL COMPATIBILITY TEST NO. 2 - 150 HOURS AT 600°F
MCS-293
25 ml Per Test Specimen

Specimen	70% Ag. 25% Polyimide -6 Tungsten Diselenide	72% Ag. + 28% Copper	70% Ag. 30% Tungsten Diselenide	Silver - S.S. Composite (480 S.S.)	Silver-Nickel Composite	Nickel Foametal Impregnated w/Ca F2 + Ba F2	Vascojet 1000	Titanium - Tin Alloy	Cobalt - Molybdenum Alloy	Plasma Flame-plated Molybdenum	Control
Weight Change	+ .0355	+ .0140	+ .0206	+ .0220	+ .0103	+ .0311	+ .0004	+ .0010	+ .0010	+ .0034	
Hardness Change	H85 to H36	F89 to F80	H94 to H98	H90 to H96.5	H83 to H82	F87 to F94	RC19 to RC11	RC36 to RC35.5	RC5 to RC5	RC39 to RC36	
Appearance of Specimen	Dulled finish on both sides.	Mating side discolored, dark deposit on opposite side.	Slightly dis- colored on both sides.	Discolored on both sides.	Brown deposit on mating side, slight darkening on opposite side,	Grey finish on both sides.	Thin dark coating on mating side, opposite side slightly dis- colored	Mating side dis- colored, no change on opposite side.	Thin dark coating on mating side, opposite side no change.	Thin dark coating on opposite side, mating side no change.	
Appearance of Mating Surface	No change on plated side, unplated side darkened.	Plated side discolored, unplated side darkened.	Plated side discolored, unplated side darkened.	Plated side slightly dis- colored, unplated side darkened.	Plated side slightly dis- colored, un- plated side darkened.	Plated side dis- colored, unplated side darkened.	Thin dark deposit on plated side, unplated side darkened.	This dark de- posit on plated side, unplated side darkened.	Thin dark deposit on plated side, unplated side darkened.	Thin dark deposit on plated side, unplated side darkened.	
Appearance of Fluid Viscosity @ 100°F CS. (25.30)	Light brown 28.50	27.30	31.10	26.90	27.50	27.55	29.70	29.50	29.40	29.70	28.7
Viscosity @ 210°F CS. (4.18)	4.89	4.45	4.87	4.35	4.30	4.46	4.58	4.71	4.54	4.56	4.44
Acid No. mg KOH/g (0.01)	0.0	.07	.11	.02	0.0	0.0	.02	.07	.11	0.0	.02

TABLE 4
FLUID - MATERIAL COMPATIBILITY TEST NO. 2 - 150 HOURS AT 600°F
MLO-60-294
25 ml Per Test Specimen

Specimen	70% Ag. 25% Polyimide +5% Tungsten Diselenide	72% Ag. +28% Copper	70% Ag. + 30% Tungsten Diselenide	Silver - S.S. Composite (430 S.S.)	Silver-Nickel Composite	Nickel Foametal Impregnated w/Ca F2 + Ba F2	Vascojet 1000	Titanium- Tin Alloy	Cobalt- Molybdenum Alloy	Plasma Flame-Plated Molybdenum	Control
Weight Change	-. 0737	+. 0003	-. 0606	+. 0241	+. 0066	+. 0012	+. 0004	+. 0004	-. 0107	+. 0015	
Hardness Change	H85 to H47	F89 to F91.5	H94 to 90.5	H90 to H84	H83 to H82	F87 to F92	RC19 to RC125	RC36 to RC32	RC5 to RC2	RC39 to RC33.5	
Appearance of Specimen	Slightly dis- colored on both sides.	Copper color on mating side, opposite side tarnished.	Slightly dis- colored on both sides.	Thin brown coating on both sides.	Thin brown coating on mating side. Tacky film on opposite side.	Dull grey on both sides.	Slight dis- coloration on mating side, opposite side darkened.	Slightly dis- colored on both sides.	No change	No change	
Appearance of Mating Surface	No change on plated side, unplated side discolored.	No change on plated side, unplated side discolored.	No change on plated side, unplated side discolored.	Brown coating on plated side, unplated side darkened.	Brown coating on plated side, unplated side darkened.	No change on plated side, unplated side darkened.	No change on plated side, unplated side darkened.	No change on plated side, un- plated side darkened.	No change on plated side, unplated side darkened.	No change on plated side, unplated side darkened.	
Appearance of Fluid	Amber	Amber	Amber	Amber	Amber	Amber	Amber	Amber	Amber	Amber	
Viscosity @ 100°F CS. -(1.5. 02)	12.75	12.90	12.55	12.40	13.00	12.80	12.90	12.75	13.10	12.75	13.1
Viscosity @ 210°F CS. (3.26)	3.04	3.02	3.08	3.04	3.06	3.05	3.07	3.06	3.09	3.00	3.05
Acid No. mg KOH/g (0.02)	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0

original viscosity of 48.18 centistokes. High acid numbers (mg KOH/g), were exhibited by the fluid samples containing the silver alloy (72% Ag, 28% Cu) and the nickel Foametal impregnated with CaF_2 and BaF_2 . Acid numbers for these fluid samples were 20.2 and 19.0 (mgKOH/g), respectively. The fluid sample containing the silver-tungsten diselenide composite material crystalized in the tube. Discussions held with the producer (Westinghouse) of this material revealed that the incompatibility may be due to an alloying element in the silver. It was also discovered that the F-50 fluid containing the Metco flame-plated molybdenum specimen solidified in the bottle (Figure 13) approximately one month after testing. This condition was believed to be caused by the evaporation of the light ends of the fluid when exposed to air during the fluid analysis work.

As experienced previously, the MCS-3101 fluid once again exhibited considerable degradation. Fluid breakdown was evidenced after 50 hours of testing when the fluid pressure in the tubes started to build up from the initial 15 psi pre-charge. At the completion of the test, pressure in the tubes containing the control sample, Polymet, silver-stainless steel composite, and Polymer SP, was 85, 85, 110, and 150 psi, respectively. The control sample was extremely viscous and black. The fluid containing the material specimens was paste-like in appearance.

2. Fluid-Material Compatibility Testing at 400°F

Fluid-material compatibility testing at 400°F was conducted with fluids that did not perform satisfactorily at 600°F. The fluids retested were F-50 silicone and MCS-3101. The MCS-3101 fluid was tested with all ten candidate seal materials. The F-50 silicone fluid which exhibited a high acidic condition when in contact with certain candidate seal materials at 600°F, was tested with only those materials in question.

Both fluids completed the 150-hour test. However, results obtained with the MSC-3101 fluid were questionable because of the removal of the bulk of the fluid from the test capsules by vaporization during the degassing process. This was not discovered until the test was completed. Normally, fluid degassing is accomplished at approximately 250°F with a vacuum of 30 inches of Hg. However, during the last three hours of the 72-hour degassing period the oven

inadvertently heated to about 375°F to 400°F. At this temperature the MCS-3101 vaporized and was drawn out of the tubes by the vacuum pump. However, degassing at this temperature did not appear to affect the F-50 silicone fluid since the fluid was intact in the tubes after testing.

Results of the silicone fluid run are summarized in Table 5. As shown in Figure 16, the fluid was slightly discolored. The seal material specimens (Figure 17) were virtually unaffected at 400°F. The high acidic condition exhibited at 600°F by the silicone fluid when in contact with the nickel Foametal (impregnated with $\text{CaF}_2 + \text{BaF}_2$), silver alloy, silver tungsten diselenide composite, and the silver-stainless steel (Type 430) composite was not present at 400°F. Variations in viscosity were also minor.

Due to the loss of the MCS-3101 fluid, there was not sufficient fluid remaining in the tubes to perform any fluid analysis. However, the material specimens appeared to be in good condition. A repeat of the 400°F run with MCS-3101 fluid has been completed, and the fluid and material specimens are now being analyzed.

3. Fluid-Material Compatibility Testing at 500°F

Fluid-material compatibility testing at 500°F was conducted with MCS-3101 and F-50 silicone fluids. The MCS-3101 fluid was evaluated with the ten candidate seal materials. The F-50 silicone fluid was evaluated with only those seal materials that indicated some incompatibility at 600°F. These were: nickel Foametal, silver-copper alloy, silver-tungsten diselenide composite, and the silver-stainless steel composite.

Results of the run made with F-50 silicone fluid are summarized in Table 6. With the exception of the silver-stainless steel composite, which exhibited slight corrosion, the F-50 fluid produced minor effects (Figure 18) on the material specimens at 500°F. The chrome-plated (440C) test buttons were in good condition except for slight corrosion on the button which mated with the silver-stainless steel composite. The high acidic condition exhibited by the F-50 fluid at 600°F was not present at 500°F. Variations in fluid viscosity were also minor. General condition of the fluid samples is shown in Figure 19.

TABLE 5. FLUID-MATERIAL COMPATIBILITY TEST NO. 3 - 150 HOURS AT 400°F

F-50 Silicone 25 ML Per Test Specimen					
Specimen	Nickel Foametal w/CaF ₂ & BaF ₂	Silver Alloy 72% Ag 28% Cu	70% Silver +30% Tungsten Diselenide	Stainless-Steel Silver Composite	Control
Weight Change (grams)	-.0019	-.0001	+.0137	+.0005	
Hardness Change (Rockwell)	From F-87 to F-91	From H-89 to H-89	From H-94 to H-99	From H-90 to H-96	
Appearance of Specimen	No change	No change	Slight discoloration	Contact side no change; opposite side corroded	
Appearance of Mating Surface	No change	No change	Slightly tarnished	No change	
Appearance of Fluid	Cloudy	Clear	Very light amber	Clear	Light amber
Viscosity @ 100°F CS (48.18)	30.59	46.83	41.78	45.18	34.98
Viscosity @ 210°F CS (15.99)	12.76	15.48	13.56	15.18	12.76
Acid No. mg KOH/g (0.03)	0.05	0.05	0	0.05	0

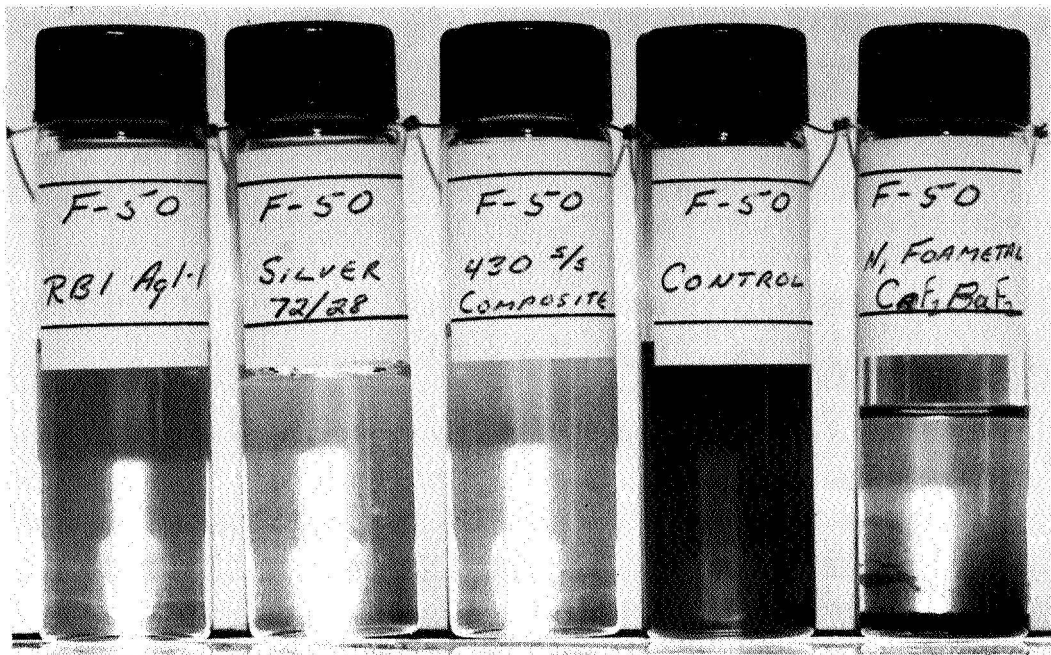


Figure 16. Test No. 3 - F-50 Silicone After Test at 400°F

Nickel Foametal	70% Ag	Silver Alloy	Silver - S.S.
Impregnated	30% Tungsten	72% Ag +	Composite
w/CaF ₂ + BaF ₂	diselenide	28% Cu	

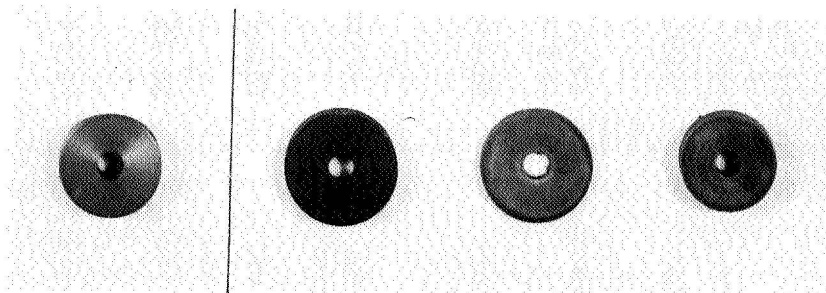


Figure 17. Test No. 3. - Material Specimens After Test at 400°F

TABLE 6. FLUID MATERIAL COMPATIBILITY TEST NO. 4 - 150 HOURS AT 500°F

F-50 Silicone Fluid
25 ML Per Test Specimen

Specimen	Nickel Foametal w/CaF ₂ + BaF ₂	Silver Alloy 72% Ag+28% Cu.	70% Silver +30% Tungsten Diselenide	Stainless-Steel Silver Composite	Control
Weight Change (grams)	-.004	-.001	+.019	-.016	
Hardness Change (Rockwell)	H-87 to H-94	F-89 to F-76	H-94 to H-80	H-90 to H-82	
Appearance of Specimen	No change	Slightly tarnished	Slightly discolored	Slight corrosion	
Appearance of Mating Surface	No change	No change	No change	Slight corrosion	
Appearance of Fluid	No change	No change	Light amber	Light amber	No change
Viscosity @100°F CS (48.18)	45.0	47.1	56.9	50.6	45.5
Viscosity @210°F CS (15.99)	14.9	15.45	17.75	16.45	15.1
Acid No. mg KOH/g(0.03)	.001	.21	.22	.20	.001

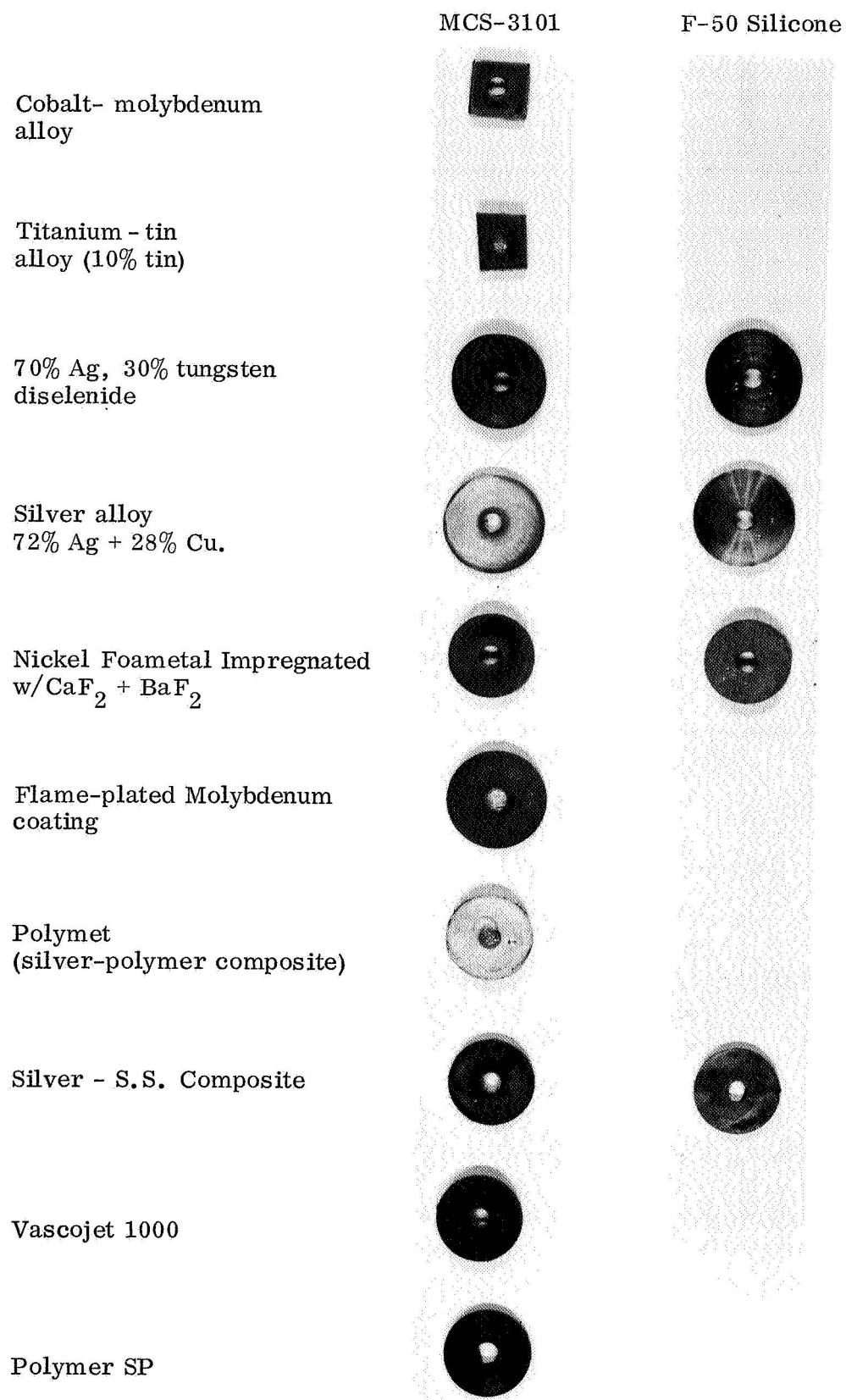


Figure 18. Test No. 4 - Material Specimens After Test at 500°F

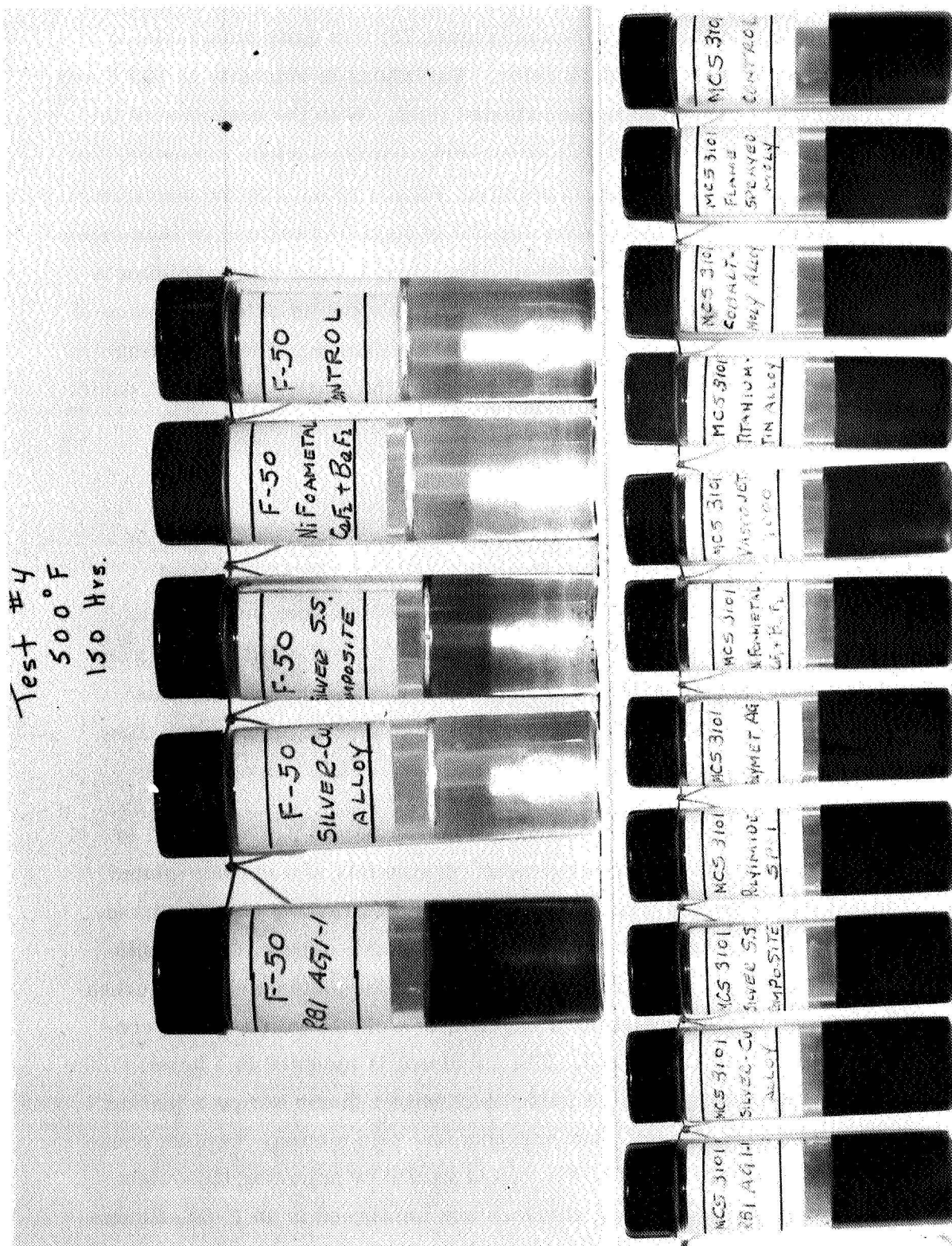


Figure 19. Test No. 4 - F-50 Silicone and MCS-3101 After Test at 500°F

Data on the run made with the MCS-3101 fluid are summarized in Table 7. Discoloration of the fluid samples (Figure 19) was quite noticeable; it ranged from dark amber to a very dark color. Variations in viscosity at 100°F and 210°F were minor when compared to the untested fluid. With the exception of the fluid sample that was in contact with the silver-tungsten diselenide composite, the fluids exhibited only nominal changes in acidity. Figure 18 depicts the condition of the material specimens. A heavy dark deposit of fluid was noticed on both sides of the silver-tungsten diselenide composite. The silver copper alloy exhibited a dull finish on the mating side and a dark coating on the opposite side. Evidence of corrosion was noticed on the silver-stainless steel composite. The remaining material specimens exhibited practically no change. The mating buttons all exhibited some coating or deposit on the unplated side.

C. SLIDING WEAR TEST

Sliding wear tests were conducted on a majority of the candidate seal materials to determine their apparent wear characteristics when subjected to reciprocating motion against a hard chrome-plated piston rod. Initially, it was intended to conduct tests only on the hard base materials because some doubt existed as to their compatibility in sliding contact with hard chrome plate. However, these tests were extended to include the soft base materials as well.

The reciprocating wear tester, shown in Figures 20 and 21, was used to simulate a reciprocating rod seal arrangement. It consists of a chrome-plated Type 440C stainless piston rod (Hardness Rc 53) and a stationary test specimen. The piston rod is supported on both ends by linear roller bearings. A variable speed motor is used to vary the reciprocating speed of the piston rod. The cross-section of the test specimen consists of a segment of a circle with a radius conforming to the radius of the piston rod. The specimen is mounted in a keyed slider that is attached to the loading arm. A guide block with a thumb screw adjustment permits vertical motion of the arm, but prevents any side motion. Contact pressure between the test specimen and piston rod is varied by adjusting the weight attached to the loading arm. Running surfaces are immersed in an F-50 silicone fluid bath. The assembly is closed off to provide a nitrogen atmosphere over the degassed silicone fluid.

TABLE 7
FLUID-MATERIAL COMPATIBILITY TEST NO. 4 - 150 HOURS AT 500° F
MCS-3101 FLUID
25 ml Per Test Specimen

Specimen	75% Cobalt 25% Molybdenum	Titanium-Tin Alloy	Silver-Tungsten Diselenide Composite	Silver Alloy 72%Ag+28%Cu	Nickel Foametal W/CaF ₂ +BaF ₂	Metco-flame plated molyb- denum	Silver-stainless steel composite	Polymer-SP Polymet	Vascojet 1000	Control
Weight change (grams)	+ .0016	+ .0022	- .1717	+ .0607	- .003	+ .0038	+ .0067	+ .0182	+ .0188	+ .0008
Hardness Change (Rockwell)	RC 5-No change	RC 36 to RC 39	H 94 to H 65	F 89 to F 69	H 87 to H 95	RC 39 to RC 36	H 90 to H 78	H 90 to H 71	H 48 to H 18	RC 19 - No change
Appearance of Specimen	No change	No change	Heavy dark deposit on both sides	Dull finish on mating side, Dark coating on opposite side	No change	No change	Corrosion on both sides	No change	No change	Slight tarnish
Appearance of Mating Surface	Greenish coating	Thin green coating on both sides	Plated side tarnished- dark deposit opposite side	Red coating on mating side, Dark coating on opposite side	Thin green coating on plated side, Opposite side tarnished	Slight etch- ing on plated side. No change on opposite side	Thin dark coating on both sides	Greenish coating	Plated side no change, Dark coat- ing on opp- osite side.	Slight etching on plated side
Appearance of Fluid	Very dark	Very dark	Very dark	Very dark	Very dark	Very dark	Very dark	Dark Amber	Very dark	Dark amber
Viscosity @ 100° F CS (4.34)	4.31	4.75	5.35	4.93	4.68	4.71	4.98	4.90	4.88	4.70
Viscosity @ 210° F CS (1.32)	1.45	1.49	1.45	1.50	1.50	1.48	1.47	1.52	1.49	1.43
Acid No. mg/KOH/g(0.01)	.02	.03	.15	.04	.02	.03	.01	.03	.02	.01
										.02
										4.81
										1.45

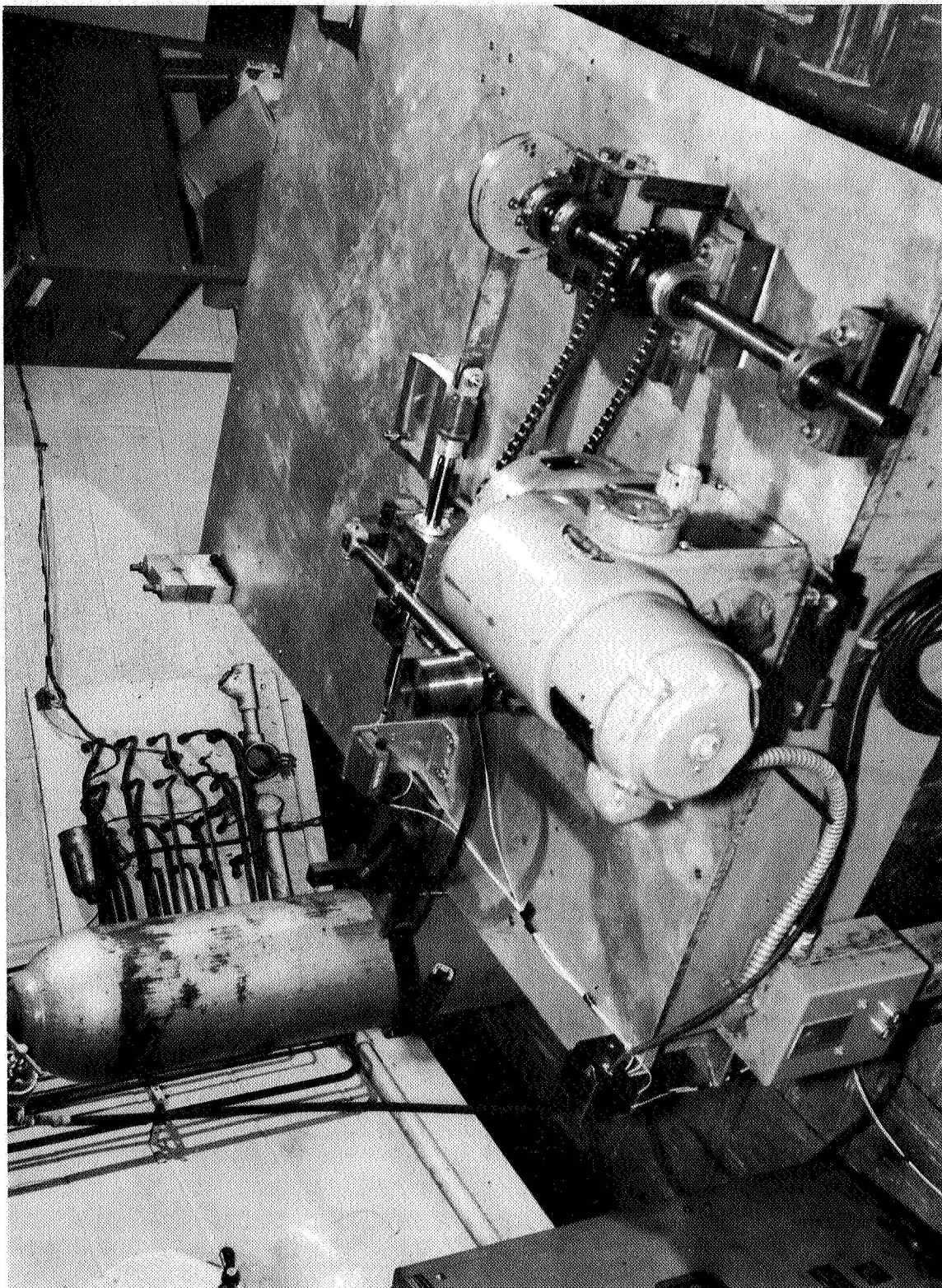
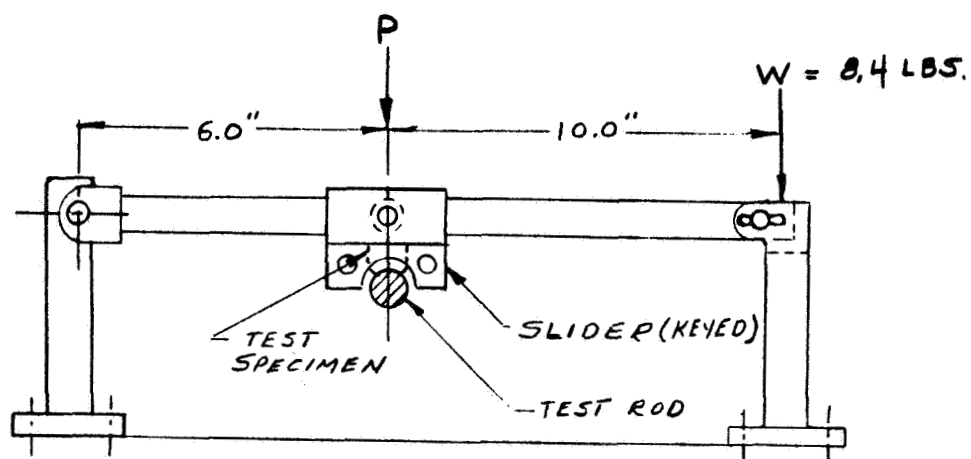
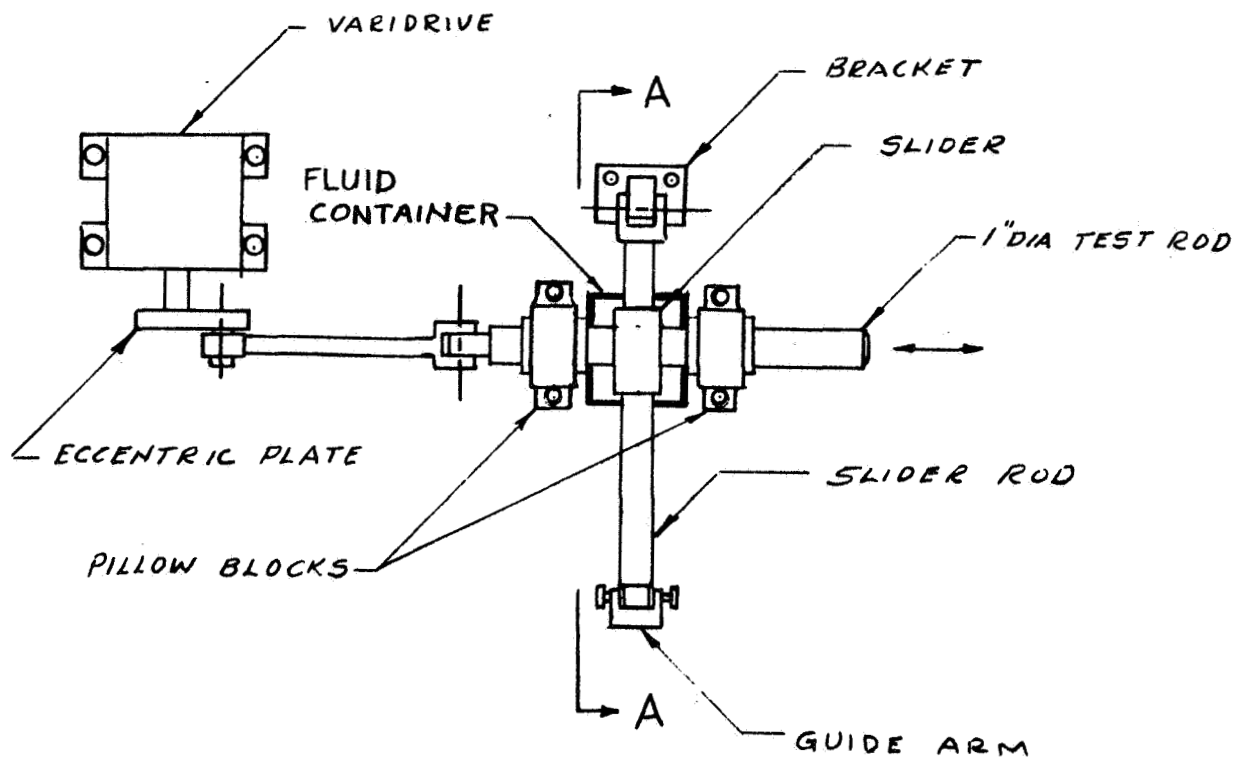


Figure 20. Sliding Wear Tester



VIEW A-A

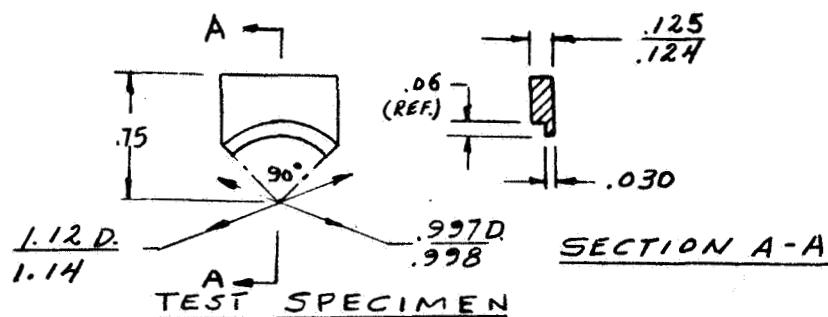


Figure 21. Details of Sliding Wear Tester

All tests were conducted on the same piston rod. The rod was shifted to a new position for each test to ensure that an untested surface was used. Testing was conducted with a piston rod stroke of .5 inch at surface speeds of .8, 1.6, and 2.5 inches per second. The material specimens were run at contact pressures of 300, 600, and 1200 psi. Failure of a specimen was defined as a two-fold increase over the initial running friction determined for a given contact load. When failure occurs, shut-down of the test is automatic. This is accomplished with a calibrated spring capsule attached to the piston rod and connected in series with the driving motor. When friction exceeds the preset load of the spring capsule, the electric contact is broken and the driving motor shuts down.

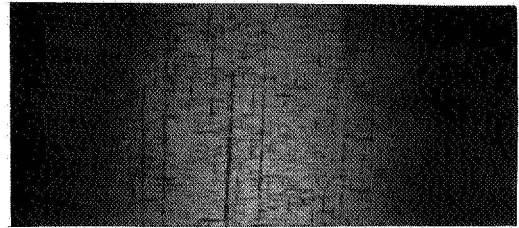
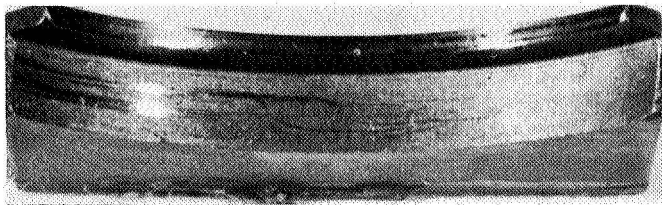
Test results are summarized in Table 8. The condition of the wear specimens and their respective mating surfaces is shown in Figures 22 and 23. In general, the results show that of the hard base materials, the Vascojet 1000 and cobalt-molybdenum alloy did not produce any significant surface damage to the chrome plating. Although the titanium-tin alloy exhibited good compatibility with chrome plate, its wear rate was rather high. The flame-plated molybdenum coating produced scratching of the chrome plating at contact pressures above 300 psi. One difficulty encountered with this material is in obtaining a good uniform surface finish. All of the soft materials tested exhibited low wear and good compatibility with chrome plating.

When examining the specimens after testing, it was noticed that they did not make full contact with the piston rod. Thus, the materials were tested under contact stresses that were approximately 25 to 50 percent higher than the calculated values.

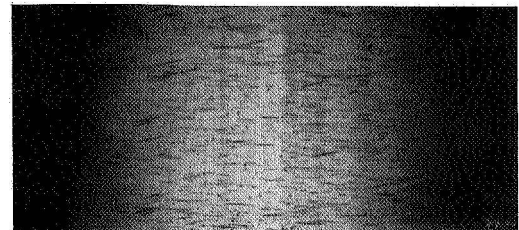
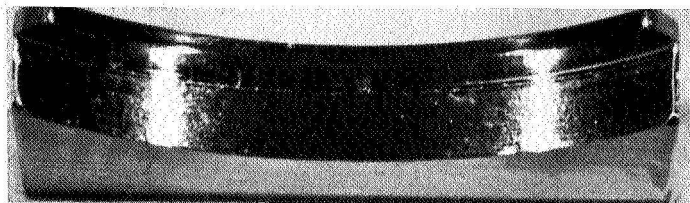
Testing of the Vascojet 1000 material was terminated after completing 147,300 inch-cycles at contact pressures up to 1200 psi. Inspection of the piston rod and specimen showed no evidence of scoring. The specimen indicated a weight loss of approximately .0011 gram. However, light burnishing marks were visible on the chrome plated surface of the piston.

TABLE 8. SLIDING WEAR TEST - SUMMARY OF RESULTS

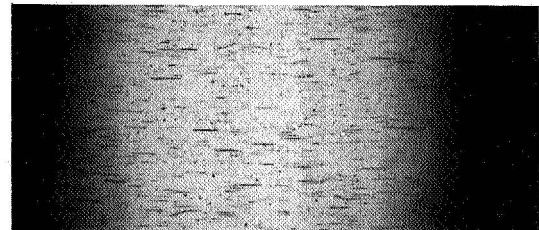
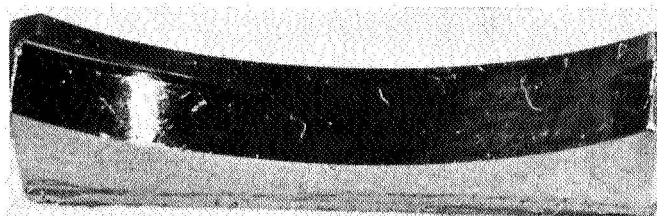
Material Specimen	Contact Pressure (psi)	Running Friction (lbs)	Surface Speed (in/sec.)	Cycles (1/2-in. Stroke)	Total Cycles	Time (hrs.)	Total Time (hrs)	Weight Loss of Specimen
Vascojet-1000	300	1.5	0.8	64,300		20		
	600	2.0	1.6	29,900		5		
	1200	2.0	2.5	53,100	147,300	6.5	31.5	.0011 gram
Titanium Tin Alloy	300	1.8	0.8	39,600		12.5		
	600	2.0	0.8	400		.5		
	600	2.0	1.6	14,300	54,300	2.5	15.5	0.131 gram
Cobalt-Molybdenum Alloy	300	1.2	0.8	31,800		12.0		
	600	2.2	1.6	64,800		11.0		
	1200	2.4	2.5	119,900	216,500	15.5	38.5	0.004 gram
Nickel Foametal w/ CaF ₂ and BaF ₂	300	2.0	0.8	39,400		12.5		
	600	2.0	1.6	47,600		10.0		
	1200	2.5	2.5	113,000	200,000	13.0	35.5	0.0017 gram
Polymet	300	1.2	0.8	38,800		12.0		
	600	1.3	1.6	53,200		9.5		
	1200	2.0	2.5	108,000	200,000	14.0	35.5	0.0004 gram
Silver Alloy (72% Ag + 28 Cu.)	300	.9	0.8	32,900		11.0		
	600	.9	1.6	57,100		10.5		
	1200	1.5	2.5	110,000	200,000	14.0	34.5	0.0004 gram
Pure Silver (Commercially Pure)	300	1.0	0.8	32,600		11.5		
	600	1.5	1.6	57,400		11.0		
	1200	1.7	2.5	110,000	200,000	12.0	34.5	0.0019 gram
Metco Flame-Plated Molybdenum (Vascojet-1000 base material)	300	2.5	0.8	39,000		13.0		
	600	3.0	1.6	52,000		8.0		
	1200	3.0	2.5	109,000	200,000	11.0	32.0	0.0008 gram



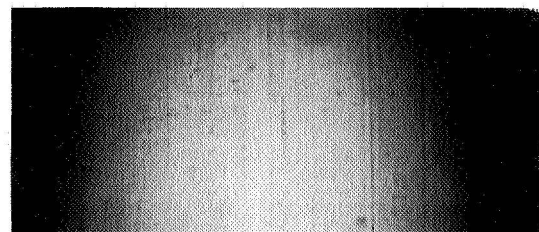
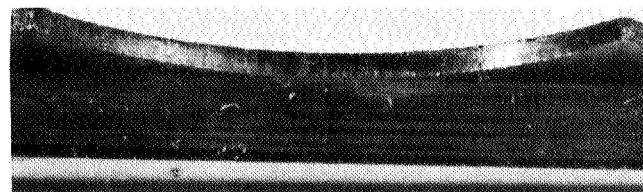
Vascojet 1000



Cobalt-Molybdenum Alloy (2% Molybdenum)



Titanium-Tin Alloy (10% Tin)

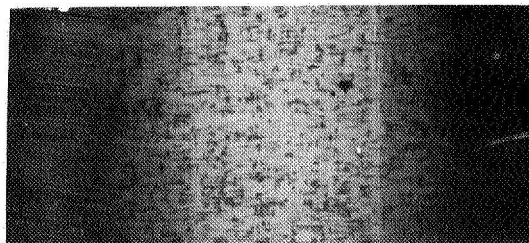
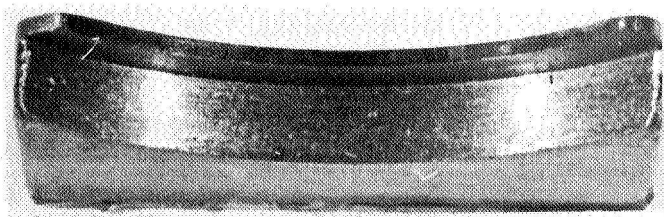


Flame-Plated Molybdenum on Vascojet 1000
(Burnished with Molybdenum Disulphide)

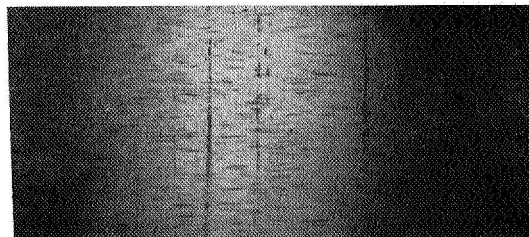
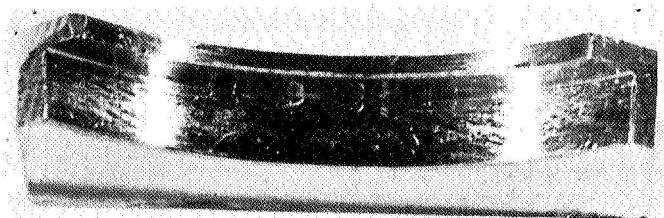
Wear Specimen 4x Magnification

Chrome Surface 25x Magnification

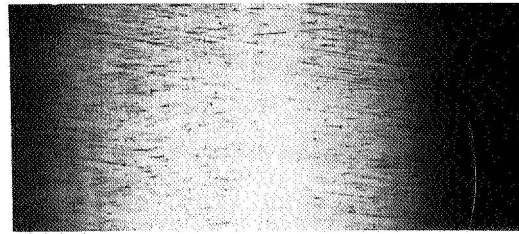
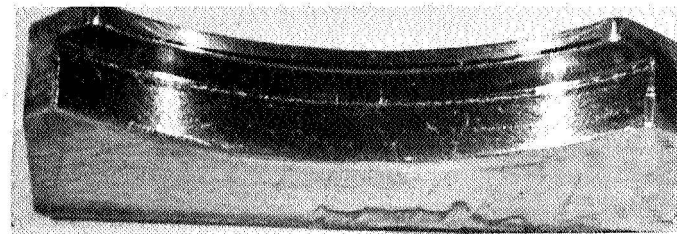
Figure 22. Sliding Wear Specimens and Chrome-Plated Mating Surfaces



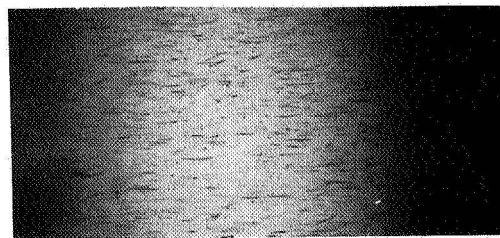
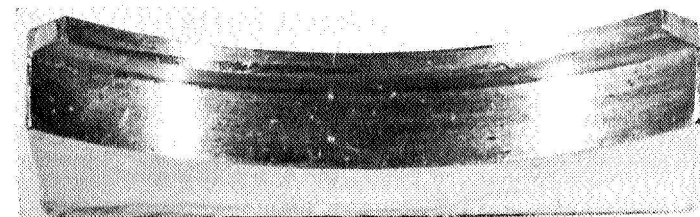
Nickel Foametal Impregnated w/ $\text{BaF}_2 + \text{CaF}_2$



Silver Alloy (72% Ag + 28% Cu.)



Silver Polymer Composite (Polymet)



Commercially Pure Silver

Wear Specimen 4x Magnification

Chrome Surface 25x Magnification

Figure 23. Sliding Wear Specimens and Chrome-Plated Mating Surfaces

Testing of the titanium alloy, which consisted of 10% tin by weight, was terminated after 15.5 hours of cycling when excessive wear was observed at the 600 psi load level. Although this material exhibited a high wear rate (approximate weight loss of .131 gram), the material did not produce any galling or other damages to the chrome plating. Hardness of the test specimen was Rc-36. The wear particles from the test specimen were dispersed in the F-50 fluid, which turned black. However, after filtering the fluid through a five micron filter paper, the fluid regained its original color.

Testing of the cobalt alloy specimen (2% molybdenum by weight) was terminated after 38.5 hours of cycling. This material, which has a hardness of Rockwell B-86, showed very good compatibility with the chrome plated piston rod. Wear rate was very low in comparison to the titanium alloy. Weight loss was approximately .004 grams after 216,500 cycles.

In the test on the flame-plated molybdenum coating, the coating was applied over a Vascojet 1000 base material and ground to an apparent finish of 4 rms. The coating was then burnished with molybdenum disulphide powder in an inerted atmosphere.

Testing of this material was terminated after 32 hours of operation. The molybdenum coating showed good wear resistance. Weight loss of the specimen was approximately .0008 gram after 200,000 cycles. Although the weight of the specimen prior to burnishing with molybdenum disulphide was not taken, it is quite conceivable that part of the weight loss is attributed to the loss of the molybdenum disulphide powder. Inspection of the piston rod revealed a highly burnished surface with a series of light longitudinal scratches. Burnishing of the surface was first noticed during cycling at a contact pressure of 300 psi. Evidence of scratching was exhibited when contact pressure was increased to 600 psi.

The nickel Foametal impregnated with CaF_2 and BaF_2 , which has a hardness of Rockwell F-87, exhibited very low wear during 200,000 cycles. Weight loss due to wear was approximately .0017 gram. Compatibility with the chrome plated piston rod was good.

The Polymet composite and the silver alloy (72% Ag + 28% Cu) exhibited very low wear after 200,000 cycles. Weight loss was .0004 gram for both materials. No wear marks were noticed on the chrome plated piston rod.

The pure silver showed a much higher wear in comparison to the silver alloy and Polymet. Weight loss due to wear was .0019 gram. The pure silver material was included in this series of wear tests for the purpose of obtaining an indication of the wear characteristic of the silver-impregnated stainless steel fiber composite. In the fabrication of seals with this composite material, a coating of silver will be retained on the contact surface of the seal to prevent the metal fibers from coming in contact with the piston rod. Therefore, the wear characteristic of the silver-metal composite is quite dependent on the wear resistance of the pure silver.

D. MECHANICAL PROPERTIES TESTING

Mechanical properties tests were conducted on a cobalt alloy (75% cobalt and 25% molybdenum). Results obtained at 600° F on two miniature tensile specimens are shown below:

	<u>Spec. #1</u>	<u>Spec. #2</u>
Modulus of elasticity, psi	30×10^6	30×10^6
Tensile yield strength (.2% offset), psi	117,100	116,842
Ultimate tensile, psi	148,950	146,315
% Elongation	1.2	1.1

Some difficulty was encountered initially in cutting the material. However, the material can be readily cut with a Buehler abrasive cut-off machine. Machining of the specimens was accomplished by using a carbide tool bit together with a slow speed and light cuts.

E. MATERIAL SELECTION

Selection of the five candidate seal materials was based on fluid compatibility, mechanical properties (Table 9), and sliding wear tests (Table 8). The materials recommended by Republic for the seal design phase (Task III), have been approved by NASA. These materials are:

- a) Polyimide plastic (unfilled)
- b) Nickel Foametal 60% Dense (impregnated with CaF_2 and BaF_2)
- c) Silver alloy (72% Ag + 28% Cu.)
- d) Vascojet 1000
- e) Cobalt molybdenum alloy (75 Co, 25 Mo)

The unfilled polyimide plastic (Polymer SP), exhibited good fluid compatibility and temperature resistant up to 600°F. The mechanical properties of the material appear adequate for seal configurations such as V-seals and lip seals. For the latter design, the rolled sheets are preferred over the billet form of the material because of their improved flexibility and higher elongation.

The nickel Foametal impregnated with an eutectic mixture of CaF_2 and BaF_2 was selected from several soft base metals. The self-lubricating potentials and good wearing qualities are desirable features of this material. Mechanical properties are also adequate for soft metal seal application. Results obtained in the fluid-material compatibility tests at 600°F indicated that the silicone fluid in contact with the Foametal exhibited a high acidic condition. However, this condition may be due to breakdown of the silicone fluid itself. Fluid compatibility tests conducted at 500°F and 400°F showed no evidence of high acidity in the silicone fluid. Compatibility of this material with PR-143AB, MCS-293, and MLO-60-294 is good.

The silver alloy consisting of 72% Ag + 28% Cu was selected from among the silver base materials. The copper content of the silver alloy provides a harder material, which should result in better wear resistance than the pure silver materials. Sliding wear tests verify this assumption. The mechanical properties of this material appear to be well suited for soft metal seal applica-

TABLE 9. PROPERTIES OF CANDIDATE MATERIALS

Mechanical Properties at 600°F	Polymer SP-1	Polymet	Silver Composite 430 SS, 30% dense	Silver Composite Nickel, * 30% dense	Consil-720 Silver Alloy 72% - 28%	Consil-600 Silver Alloy, * 1000 (H-11) 60% - 40%	Vascojet-600 (H-11) 60% dense	Nickel Foametal Impregnated w/Ca F ₂ and Ba F ₂
Tensile Modulus of Elasticity (psi)	15.0 x 10 ⁴	1.44 x 10 ⁵	5.47 x 10 ⁶	10.8 x 10 ⁶	6.0 x 10 ⁶	7.5 x 10 ⁶	23.9 x 10 ⁶	9.07 x 10 ⁶
Tensile Yield Strength .2% offset (psi)	1285	3444	11,525	12,277	21,325	30,997	183 x 10 ³	8297
Ultimate Tensile (psi)	1325	5805	18,762	19,333	32,550	41,361	225 x 10 ³	8297
% Elongation	3.25%	7.6%	8.7%	11.5%	15.8%	24.7%	1.8%	2.45%
Hardness**	Rockwell H-83-89	Rockwell H-53	Rockwell H-90	Rockwell H-83	Rockwell F-89	Rockwell H-97	Rockwell C-52	Rockwell F-87
Coefficient of Friction	*** 0.08-0.15	*** 0.06-0.14						
Thermal Coefficient of Expansion (in./in./°F)	29.8 x 10 ⁻⁶	8.9 x 10 ⁻⁶					6.8 x 10 ⁻⁶	

* Alternate material

** Hardness readings obtained at room temperature

*** Vendor's data

tions. Results obtained in the 600°F fluid compatibility test indicated that the F-50 silicone fluid became quite acidic after being in contact with this material. However, partial breakdown of the silicone fluid may have been a contributing factor. The acidic condition of the fluid was considerably lower at 500°F and negligible at 400°F. Its compatibility with the PR-143AB, MCS-293, and MLO-60-294 fluids was good.

Vascojet 1000 is a tool steel type material that is suited for hard metal seal application. This material is capable of retaining high strength at elevated temperatures. It has also demonstrated low wear and fairly good compatibility in sliding contact with chromium plating.

Cobalt-molybdenum alloy (75% cobalt, 25% molybdenum). A sliding wear test conducted on a cobalt-molybdenum alloy of similar composition indicated low wear and good compatibility with chromium plating. Compatibility with fluids is good. High temperature mechanical properties of this alloy appear quite suitable for hard metal seal application where good tensile properties are required.

Among the materials not selected was the silver-tungsten diselenide composite, which was grossly incompatible with the F-50 silicone fluid at 600°F. The silicone fluid in contact with this material crystallized. Another material not considered was the titanium tin alloy. This alloy exhibited excessive wear when in sliding contact with chromium plating. Although the Polymet material exhibited very low wear in the sliding wear test, it was not considered because of the organic filler used. The filler is a fluorocarbon type polymer which may tend to partially decompose during long exposure to elevated temperatures, resulting in a porous silver structure.

The silver-impregnated stainless steel (Type 430) fiber composite exhibited some corrosion effects when in contact with the F-50 silicone fluid. Therefore, it was dropped as a candidate material. The pure silver impregnant also exhibited higher wear than the silver-copper alloy.

The Metco flame-plated molybdenum coating which exhibited good compatibility with chrome plating under low load conditions, may be considered later in the program as a wear resistant coating for hard metal seals.

Procurement of the candidate materials has been initiated. To date, the Polymer SP, silver-copper alloy, Vascojet 1000, and nickel Foametal have been received. The latter material has been sent to NASA to be impregnated with calcium fluoride and barium fluoride. The cobalt-molybdenum alloy is being obtained by NASA.

TASK III - SEAL DESIGN AND DEVELOPMENT

A. GENERAL

Efforts in this phase were directed towards further development of concepts for the second-stage seal, preliminary seal evaluation, and selection of the best seal concepts. Detail design and preliminary testing of the selected seal designs have been initiated. Evaluation of seal concepts for the first-stage seal is also underway.

B. SEAL DESIGNS

Included in the seal design work was a review of the rod seal configurations investigated by Republic in past programs. Those configurations that were tested with some degree of success were included as candidate seal designs, and are briefly described below.

Figure 24 depicts a reed type seal which consists of a series of thin walled elements preloaded on an assembly to provide positive contact with the piston rod. Limited testing at 600°F and 3000 psi indicated that effective sealing can be obtained. However, rapid wear of the seal and galling of the chrome plated piston rod were major problems.

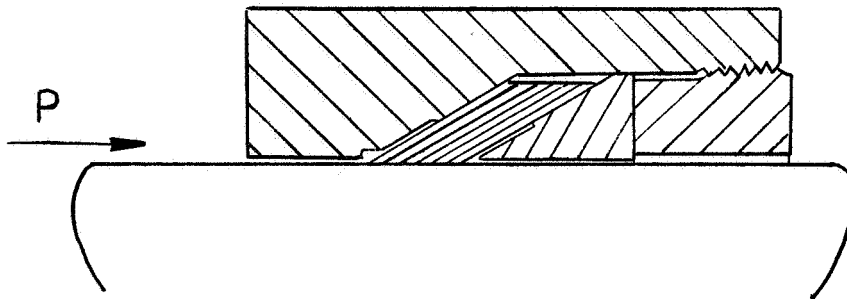


Figure 24. Metallic Reed Seal

The seal shown in Figure 25 consists of a wedge-shaped ring held in close contact with the piston rod by an axial force supplied by spring washers. The spring load also acts to compensate for wear. This configuration (Ref. 4) gave satisfactory results up to 800°F and 4000 psi for short periods. Best results were obtained using graphite and the silver-stainless composites as the seal materials (Ref. 5).

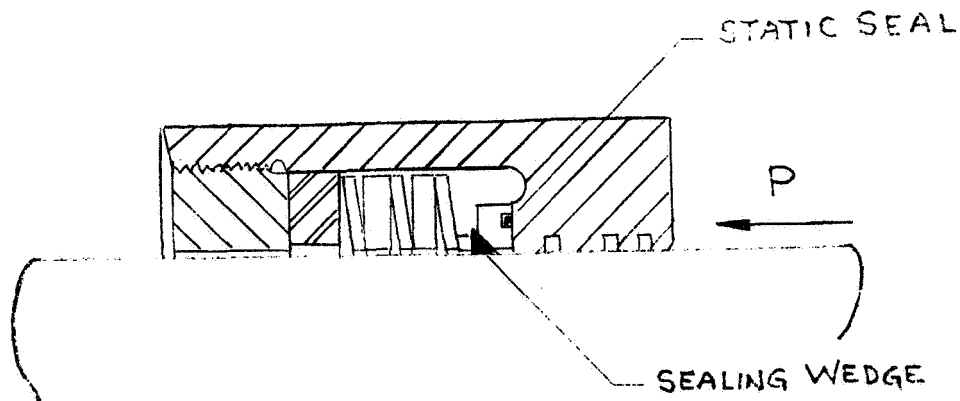


Figure 25. Spring Loaded Wedge Seal

The ring spring seal shown in Figure 26 consists of two outer rings and one inner ring. Sliding of the two outer rings occurs along the tapered surfaces of the inner ring upon application of an axial load, resulting in compression of the inner ring against the piston rod. This configuration has been evaluated in the 600°F to 900°F range at 4000 psi (Ref. 5). Seal life is relatively short due to low wear compensation.

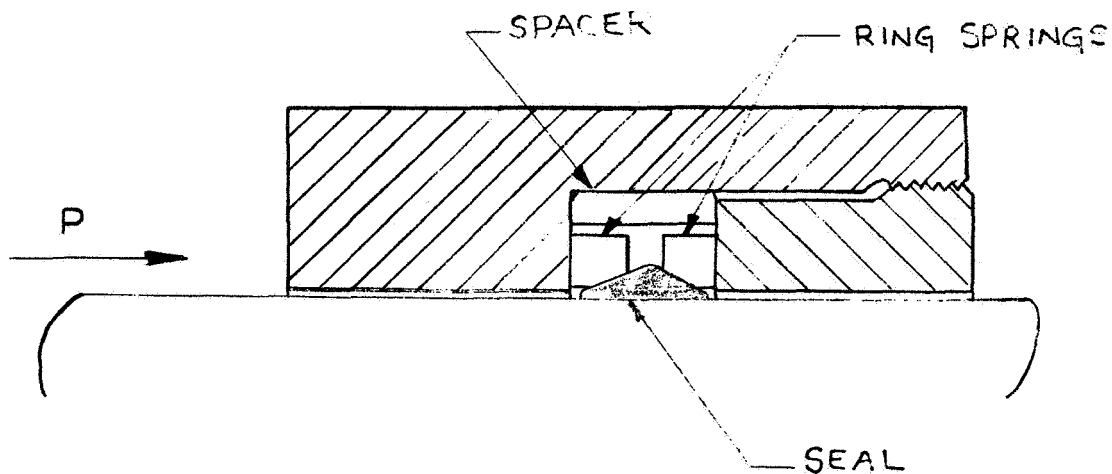


Figure 26. Metallic Ring Spring Seal

Figure 27 depicts a design consisting of an X-shaped seal and two loading rings. Sealing is accomplished by an axial force applied to the load rings, which in turn deflect the legs of the X into intimate contact with the piston rod; in a similar manner, a seal is also effected on the outer diameter of the gland to seal statically. Lack of wear compensation was the major shortcoming of this design.

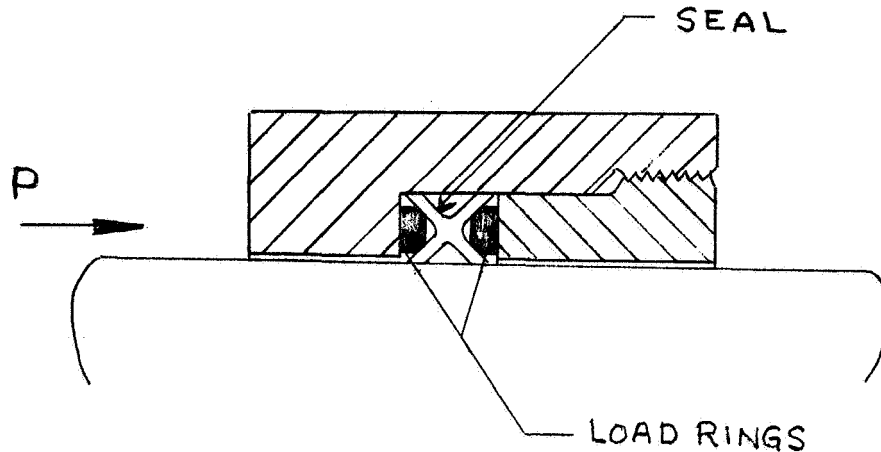


Figure 27. Metallic X-Seal

The C-shaped seal shown in Figure 28 utilizes a slight preload to provide initial contact at the static surface and between the seal and piston. The seal is also pressure-energized during operation. Seals fabricated from the silver-stainless composite have been tested up to 1000°F. Satisfactory performance was obtained at 100 psi (Refs. 3 and 5).

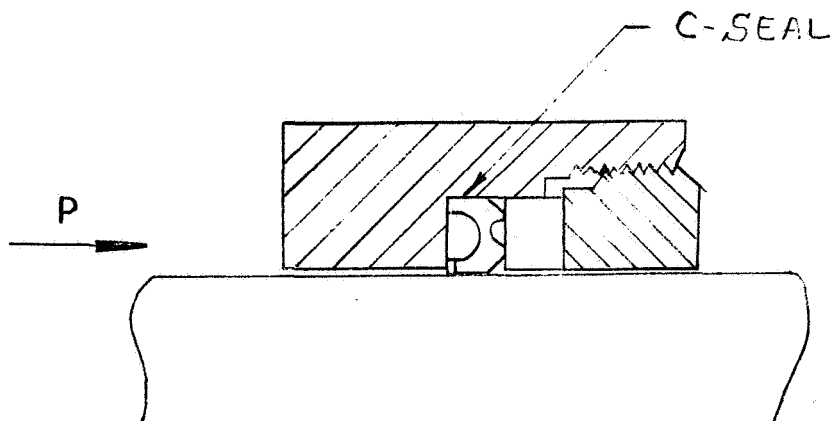


Figure 28. Metallic C-Seal, Pressure Energized

Figure 29 depicts a metallic lip seal currently under development by Republic. The design of the seal provides for an interference fit over the rod to effect a seal. In order to avoid a line contact condition, which would result in high contact stresses at the seal interface, the design provides a contact width of approximately .030 to .040-inch on assembly. This enables the sealing load to be distributed over a relatively large area, which results in lower unit contact stresses. By having the sealing lip facing away from the fluid, excessive build-up of contact stresses due to the pressure of the fluid is avoided. This arrangement also permits relieving of the sealing load at the seal interface since the fluid pressure will tend to open the inner diameter of the seal. Such a feature could be advantageous in reducing friction and wear, providing that leakage does not become excessive at high pressure.

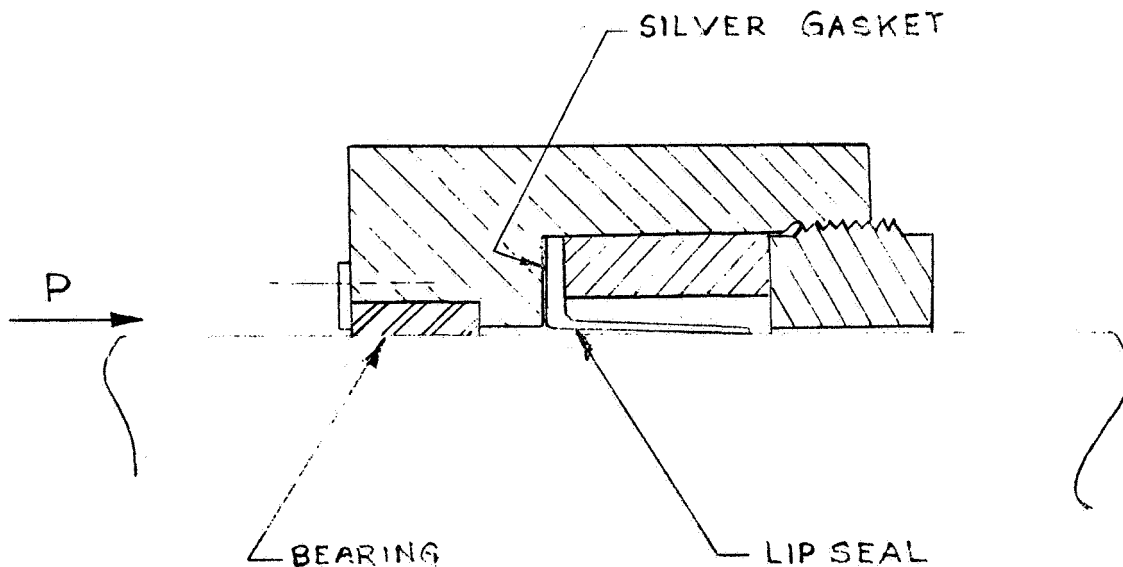


Figure 29. Metallic Lip Seal

C. PRELIMINARY SEAL EVALUATION

The metallic lip seal shown in Figure 29 was fabricated and tested to determine friction, leakage and wear under dynamic cycling. The seal, fabricated of Vascojet 1000, was intended to provide an initial interference fit of .0019-inch, which would result in a contact stress of approximately 1200 psi. However, the actual interference was approximately .0015 inch due to dimensional changes during the heat treating process. Although assembly of the seal onto the rod was relatively easy, several light scratch marks were visible on the chrome-plated surface after assembly.

Seal leakage and friction measurements were made in the test fixture (Figure 30) at room temperature with F-50 silicone fluid. During an attempt to obtain leakage and friction data at higher pressures, friction values became erratic. This condition was attributed to the O-ring that was installed on the opposite end of the seal tester. The O-ring friction generated by the higher pressures obscured the frictional characteristics of the metal lip seal. Consequently, the O-ring was removed from the tester and replaced with a metal lip seal. The interference fit of the second lip seal was somewhat higher (.0019 inch) as compared to the first seal (.0015 inch). With the installation of the second lip

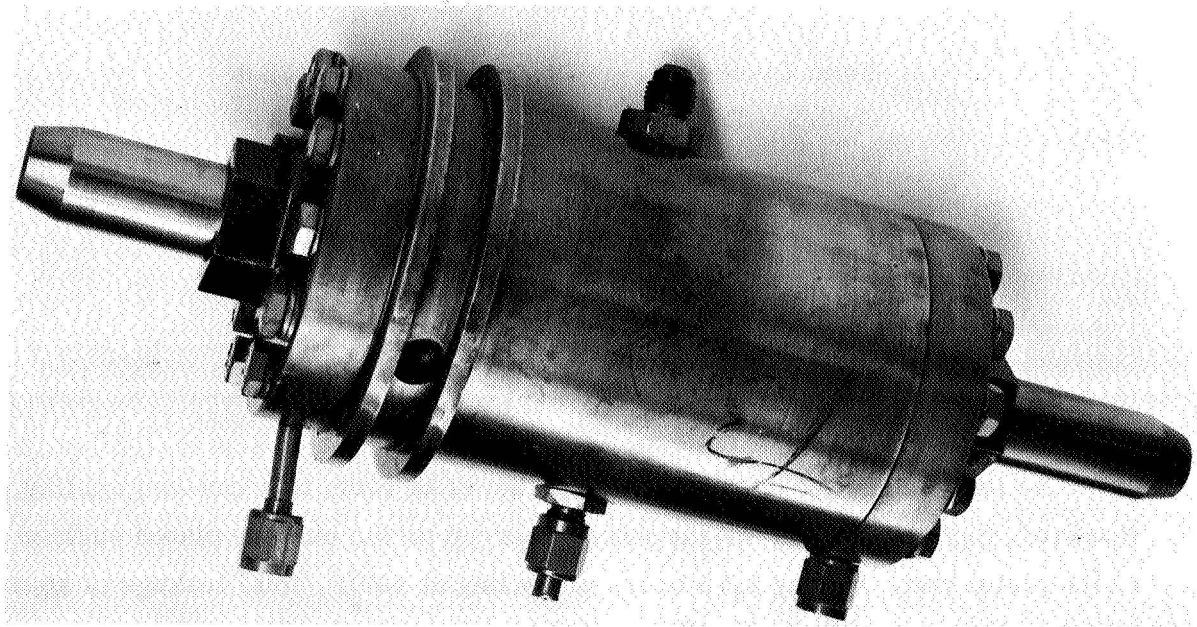


Figure 30. Seal Test Fixture

seal, the friction and leakage measurements were rerun. The data obtained are shown in Figure 31; seal breakaway friction decreased quite rapidly with increasing pressure. Seal breakaway friction began to level off to a constant value at pressures between 1500 and 2000 psi. Part of the friction indicated for this pressure range may be attributed to friction of the close fitting graphite bearings in the tester. It was also noticed that at the point where friction leveled off, leakage was observed from the "A" seal (which had a lower interference fit). Leakage started at approximately 8 drops per minute at 1500 psi and increased to 75 drops per minute at 2000 psi.

Upon completion of the above investigation, the "B" seal (see Figure 31) was removed from the tester, replaced with an O-ring seal, and retained for future investigations. The "A" seal was subjected to mechanical cycling at room temperature and at 400°F to determine the behavior of the Vascojet 1000 material sliding on chrome plate and the leakage characteristics of the seal under dynamic conditions. During the cycling, the seals were pressurized to 100 psi. The piston rod was cycled at a rate of 40 cpm with a stroke of 1.5 inches.

The time and cycles accumulated during this test are shown below:

Lip Seal Cycling Test

Total Cycles	34,200
Cycles - Room Temp.	18,000
Cycles - Room Temp. to 400°F	9,600
Cycles - 400°F	6,600
Total time, hrs	15.5
Time - Room Temp., hrs	7.5
Time - Room Temp. to 400°F, hrs	4.25
Time - 400°F, hrs	2.75

No leakage was observed during 7.5 hours of room temperature cycling. However, light burnishing marks were observed on the chrome plated surface of the piston rod. During 2.75 hours of cycling at 400°F, total leakage of approximately .25 cc was collected. This leakage represents the film of fluid that was

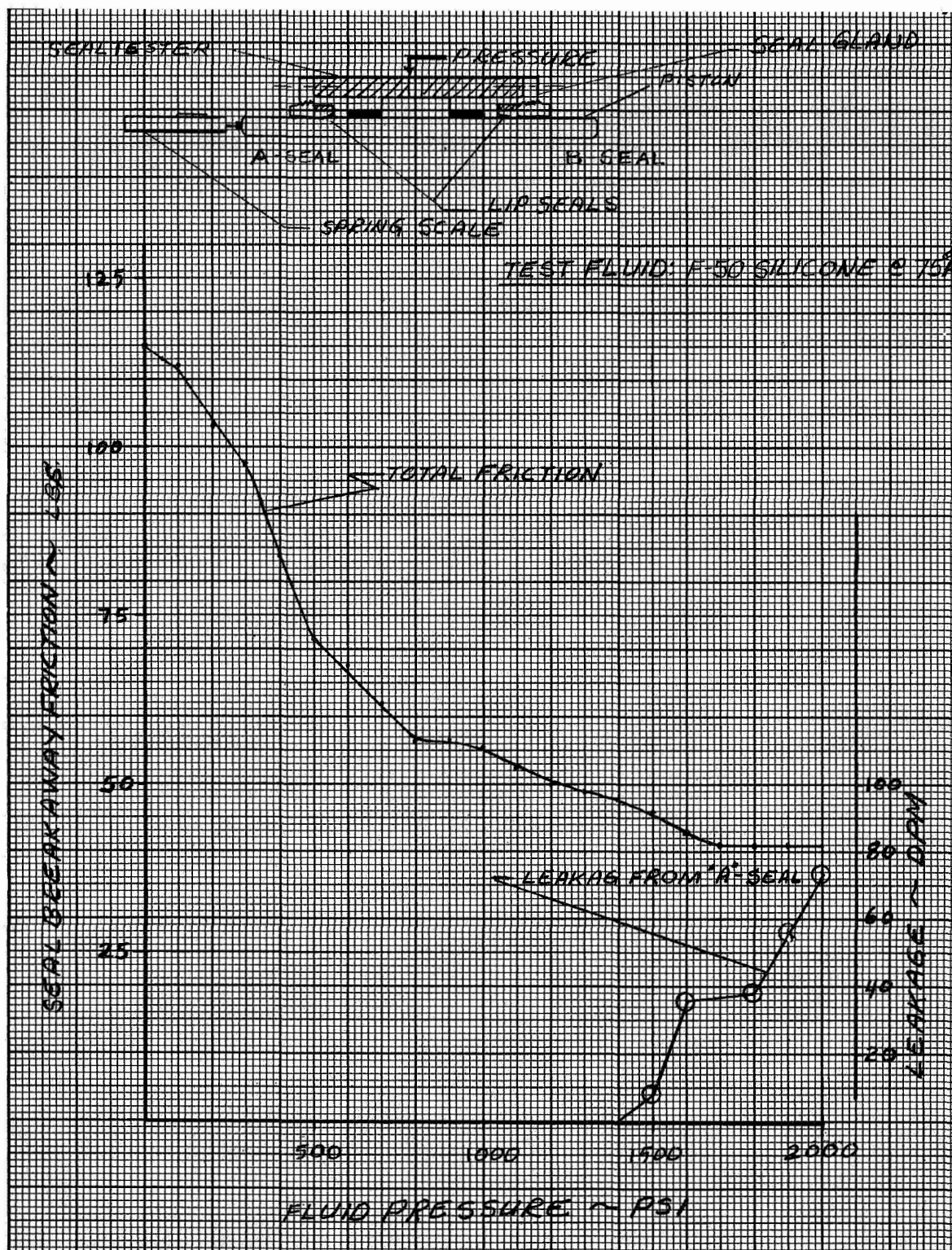


Figure 31. Seal Friction and Leakage vs Pressure
 Vascojet 1000 Lip Seal - .009 Inch Seal Thickness

carried out on the surface of the piston rod.

Test results obtained on the lip seal design indicate the feasibility of this approach to a hard metal seal. The leakage and friction data indicate that the seal contact load can be relieved by pressure. However, this configuration permits load relief to take place only at the higher pressures. Since the seal is intended for use as a second-stage low pressure seal, pressure relief is more desirable in the lower pressure range.

Based on the above findings, the metal lip seal was redesigned to obtain lower friction. This was accomplished by reducing the thickness of the seal from .009 to .0055 inch. As a result, the radial loading was reduced from 50 to 25 pounds per inch. This produced a contact pressure at the seal interface of approximately 600 psi. Seal interference (diametral) was calculated to be approximately .004 inch.

The redesigned seal was tested with F-50 silicone fluid in the seal test fixture to determine friction and leakage characteristics. Figure 32 indicates a substantial reduction in friction as compared to the original design. The actual diametral interference was .0015 and .0012 inch for the A and B seal, respectively, which was considerably less than the calculated interference of .004 inch. This difference was caused by the seals being machined oversize at the inner diameter. Leakage of 1/2 drop per minute from the B seal was first noticed at 600 psi and increased to 14 drops per minute at 800 psi.

Preliminary tests were conducted on the externally loaded lip seal shown in Figure 33, to determine the feasibility of this design approach. The lip seal was fabricated from a silver alloy (72% Ag + 28% Cu) with a seal section thickness of .025 inch. The diametral clearance between the seal and piston rod (chrome plated (Type 440C) was approximately .002 inch. Loading of the sealing lip to achieve contact with the piston rod was readily accomplished. Seal contact with the rod was indicated by an increase in seal breakaway friction (approximately 40 pounds at 0 psi). During the initial loading of the seal, leakage was approximately 5 drops per minute at 100 psi using the F-50 silicone fluid. This leakage was due to the seal not making sufficient contact with the piston rod. Consequently, the seal was run-in for 8400 cycles (30 cpm at 2-inch rod strokes)

SEAL THICKNESS. .009-IN. AND .0055-IN.
SEALS TESTED IN PAIRS

SEAL INTERFERENCE		
SEAL THKNSS	.009	.0055
A-SEAL	.0015	.0015
B-SEAL	.0019	.0012

TEST FLUID: F-50 SILICONE @ 75°F

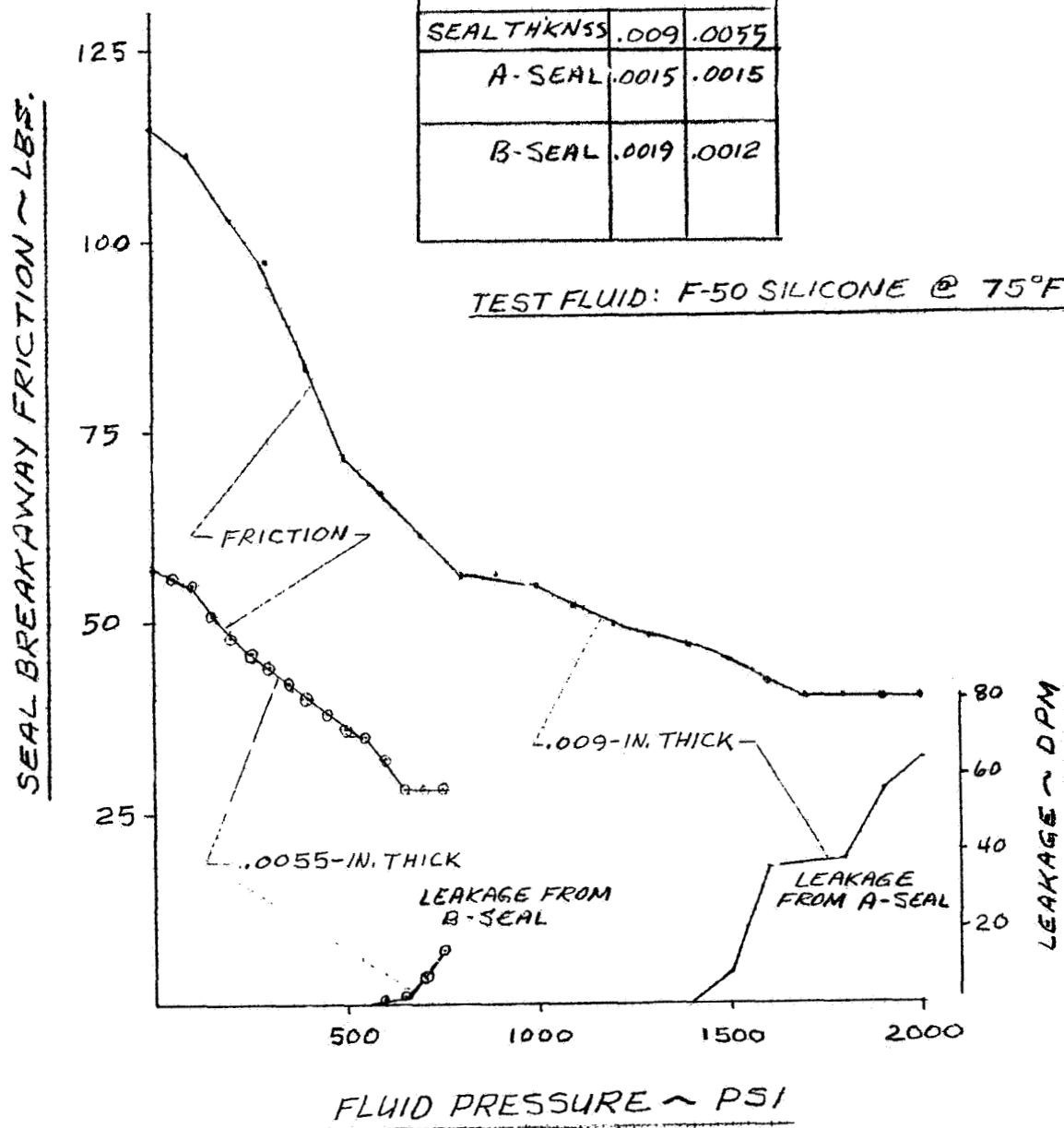


Figure 32. Seal Friction and Leakage Versus Pressure
Vascojet 1000 Lip Seal - .0055-Inch Seal Thickness

and the gland nut was retightened. Following this procedure, a pressure check of the seal was made at 100 psi with no leakage.

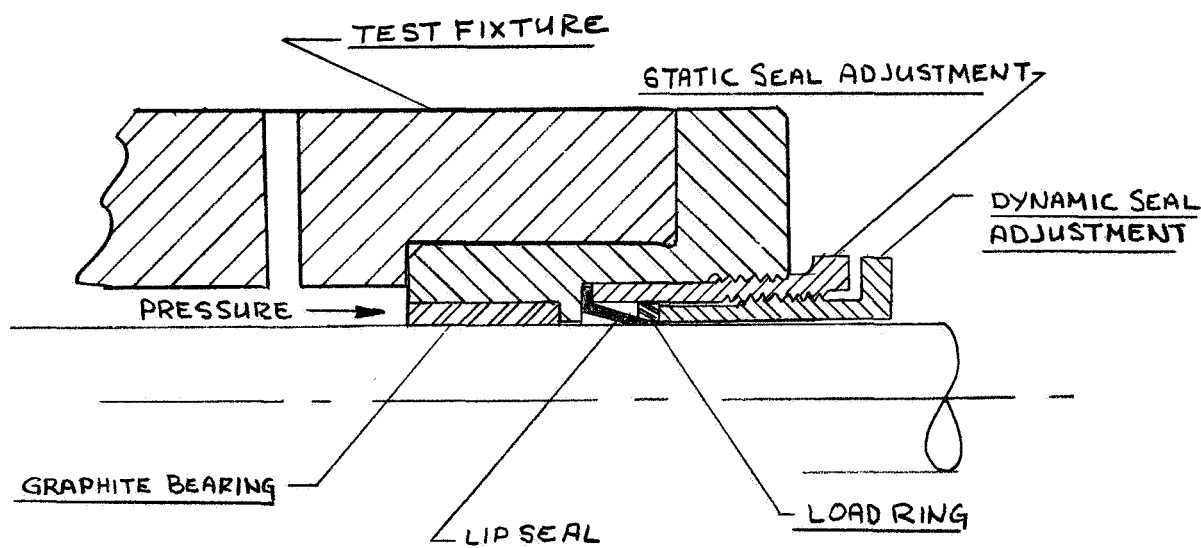


Figure 33. Lip Seal - Externally Loaded

After demonstrating sufficient sealing ability, the seal was subjected to cycling at temperatures to 400°F. Leakage collected during 6.5 hours of cycling was 12 cc. Time and cycles are summarized below.

Run-in cycles	8,400	-	4 hrs
Cycles - room temp. to 400°F	4,200	-	2 hrs
Cycles at 400°F	9,450	-	4.5 hrs
Total cycles and time	22,050	-	10.5 hrs

Disassembly of the seal shows that an even wear pattern was obtained around the sealing surface. Although the design was intended to be used with spring washers for wear compensation, no springs were used in this test since the main purpose was to demonstrate the loading approach.

D. SELECTION OF CANDIDATE SEAL DESIGNS

In selecting the most promising concepts for the second-stage seal, a rating system was established. The philosophy, basic approach, rating work

sheets, and summary of the rating results are discussed in detail in Appendix A.

In brief, the rating system provided a means of selecting seal designs that warrant further development effort. Candidate designs were rated on the following factors deemed essential to attain program objectives.

1) Sealing mechanism - Seal conforms to rod by means of a feasible loading technique. Appropriate stress levels and patterns are generated in the seal element throughout the temperature and pressure range. Geometry and behavior of elements can be predicted and controlled within workable limits. Effective static sealing is available. Pressure does not increase leakage.

2) Wear and compensation - Seal exhibits either a low wear rate or the ability to compensate for wear, and wears rod surface at a low rate. Wear process is not detrimental to overall performance of seal installation (i.e., no adverse effect on static sealing). Pressure does not increase wear.

3) Reliability - Seal fails slowly, observable over a period of time. Rugged construction, inherently resistant to abuse in use. Redundant sealing elements. Precise definition of operating conditions and behavior is not critical; possesses inherent latitude to withstand pressure surges.

4) Short term potential - Knowledge of concept is available or can be acquired in the near future. Configuration can be optimized and evaluated within time period of program.

Seal designs that achieve a high score on the above factors are then rated on the basis of other factors, considered desirable but not absolutely essential, to aid in the final selection. These desirable factors are:

1) Design features - Materials exhibit compatible coefficients of expansion. Rod does not require exotic plating. Friction is relatively low (for good servo performance). Seal has a relatively high degree of self-compensation for wear. Geometry lends to pressure balancing or pressure relief.

2) Cost - Relatively easy to manufacture (reasonable tolerance requirements, accessible geometry for machining). Materials are available and machineable.

3) Serviceability - Easy to install and remove. Does not require custom tailoring; no special installation tools required.

4) Past experience with similar approach.

Approximately 17 designs (see Appendix A) were evaluated with the rating system. The seal designs selected and their applicable materials are shown in

Table 10. Of the designs selected the following were approved by NASA for further development.

- 1) Design B, V-seal with Polymer SP material
- 2) Design I, wedge seal with nickel Foametal
- 3) Design AH multiple reed seal with either Vascojet 1000 and silver-copper alloy or cobalt molybdenum alloy and silver-copper alloy.

Seal design D lip seal (refer to Table 10) with Vascojet 1000 and cobalt molybdenum alloy will be approved as the two remaining seal designs providing the following feature can be incorporated in the design.

- 1) Fail-safe backup ring for high pressure conditions.
- 2) Flexibility in the lateral direction to accommodate a limited amount of bearing wear and lateral motion.

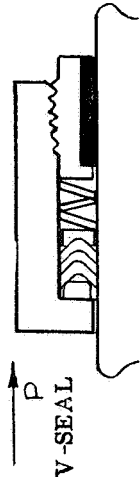
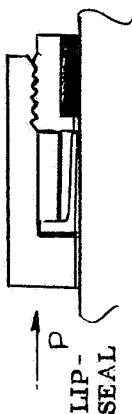
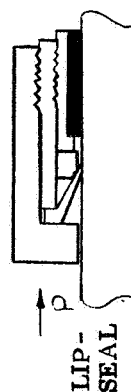
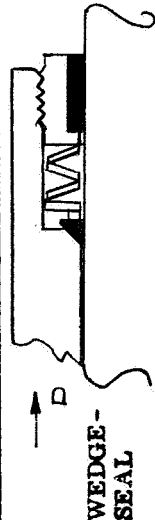
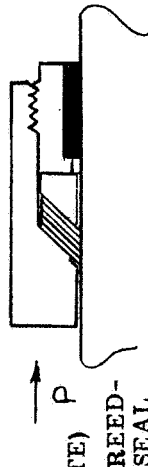
E. DESIGN AND EVALUATION OF CANDIDATE SEALS

Detail design of the Polymer SP V-seal (design B, Table 10) has been initiated. Sample seals have been fabricated and load-deflection tests are being conducted to determine the ability of the seal to compensate for wear. These data will also be used in the design of the loading springs. The test setup, depicted in Figure 34 consists of a seal gland, dial indicator, and a loading cylinder. A single V-seal is assembled in the gland with a load ring and backup ring. The dial indicator, which measures the lip deflection, is placed at the outer edge of the lip. Axial loading is applied to the seal by the loading cylinder. Data obtained to date (Figure 34) indicate that with the present design, a radial deflection of .0038 inch can be obtained with an axial load of 140 pounds.

Since multiple seal elements will be used in the final configuration, additional tests will be conducted to determine lip deflection with the seals stacked. Tests will also be conducted to determine the effects of side motion of the piston rod on the seal.

Investigations were conducted to determine the effects of bearing wear and side motion on the metallic lip seal (design D). These conditions were

TABLE 10. RECOMMENDED SEAL DESIGNS AND MATERIALS

SEAL DESIGN	SEAL TYPE	RECOMMENDED MATERIALS	ALTERNATE MATERIALS
B	 V-SEAL	Polymer SP	Nickel Foametal with $\text{Ca F}_2 + \text{Ba F}_2$
D	 LIP-SEAL	Vascojet-1000	
D	SAME	Cobalt-molybdenum alloy (75% Co + 25% Mo)	
F	 LIP-SEAL	Silver alloy (72% Ag + 28% Cu)	Nickel Foametal with $\text{Ca F}_2 + \text{Ba F}_2$ polymer SP
I	 WEDGE-SEAL	Nickel Foametal with $\text{Ca F}_2 + \text{Ba F}_2$	Silver alloy (72% Ag + 28% Cu)
(ALTERNATE) H or AH	 REED-SEAL	Vascojet - 1000 and silver alloy (72% Ag + 28% Cu) combination	Cobalt-molybdenum alloy and silver alloy (72% Ag + 28% Cu) combination, Polymer SP

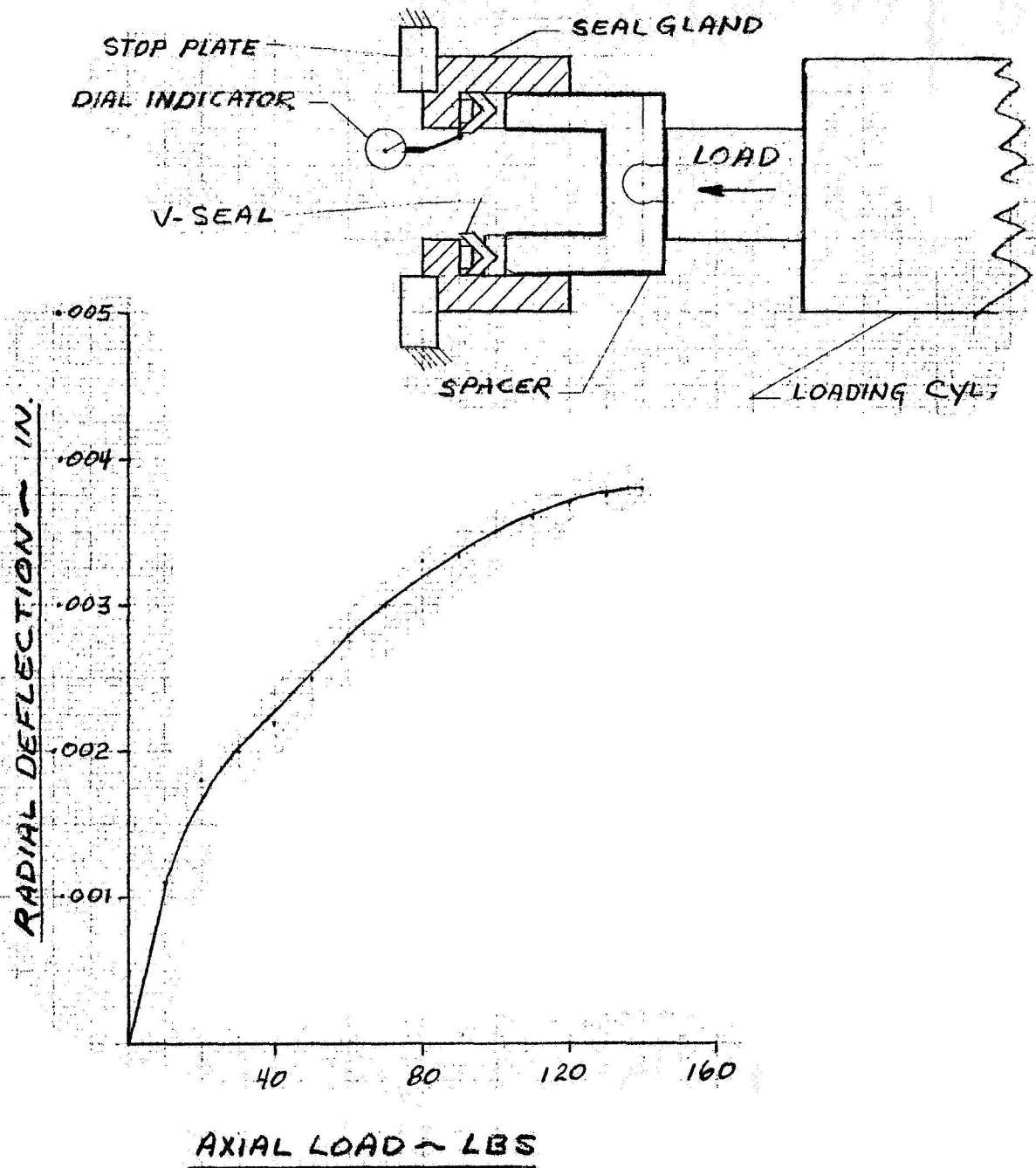


Figure 34. Polymer SP - V-Seal Test
Radial Deflection versus Axial Loading

simulated with the test setup shown in Figure 35. The rod bearings were placed inboard of the seals. Side motion on the rod was induced by use of Nylon screws fitted on the end plugs. A dial indicator was placed on the piston rod (3.765 inches from the contact point of the seal) to measure the lateral displacement of the rod. The seal was pressurized to 100 psi using benzene as the test fluid. Benzene was selected because its low viscosity (.7 centistoke at room temperature) is simulative of a hydraulic fluid at 500°F to 600°F.

Tests were conducted on seals having cross-sectional thicknesses of .009 inch and .0055 inch. In the testing of the .009-inch thick seal, the rod bearings were machined to provide a diametral clearance of .0025 inch. Bearing clearance for the .0055-inch thick seal was .004 inches. To simulate a condition wherein wear has taken place on the seals as well as the bearings, the seals tested had an interference fit of .0012 and .0015 inch for the .009- and .0055-inch thick seals, respectively. (Design interference is .0038 to .004 inch.)

Results (Figure 36) indicate that the thinner section seal was able to withstand greater lateral motion of the piston rod. For the .009 inch seal a leakage of 2 drops per minute was first observed when the dial indicator read .005-inch deflection. This corresponded to a deflection at the seal of .00275 inch. Leakage increased rapidly with increasing deflection. Leakage on the .0055-inch thick seal did not occur until a deflection at the seal of .0041 inch was induced. A rapid increase in leakage was also noticed with a further increase in deflection.

Calculated bearing clearances required to induce the above seal deflections were .00238 inch and .0036 inch for the .009- and .0055-inch thick seals, respectively. A comparison was also made between the test configuration (Figure 35) and the configuration shown in Figure 37, which has the bearings located outboard of the seal. To obtain the equivalent seal deflection with outboard bearings, bearing clearances of .00328 and .00493 inch are required for the .009-inch and .0055-inch thick seals, respectively. These are calculated values.

Since bearing clearance represents bearing wear, bearings located outboard of the seals would have a tendency to tolerate somewhat greater wear.

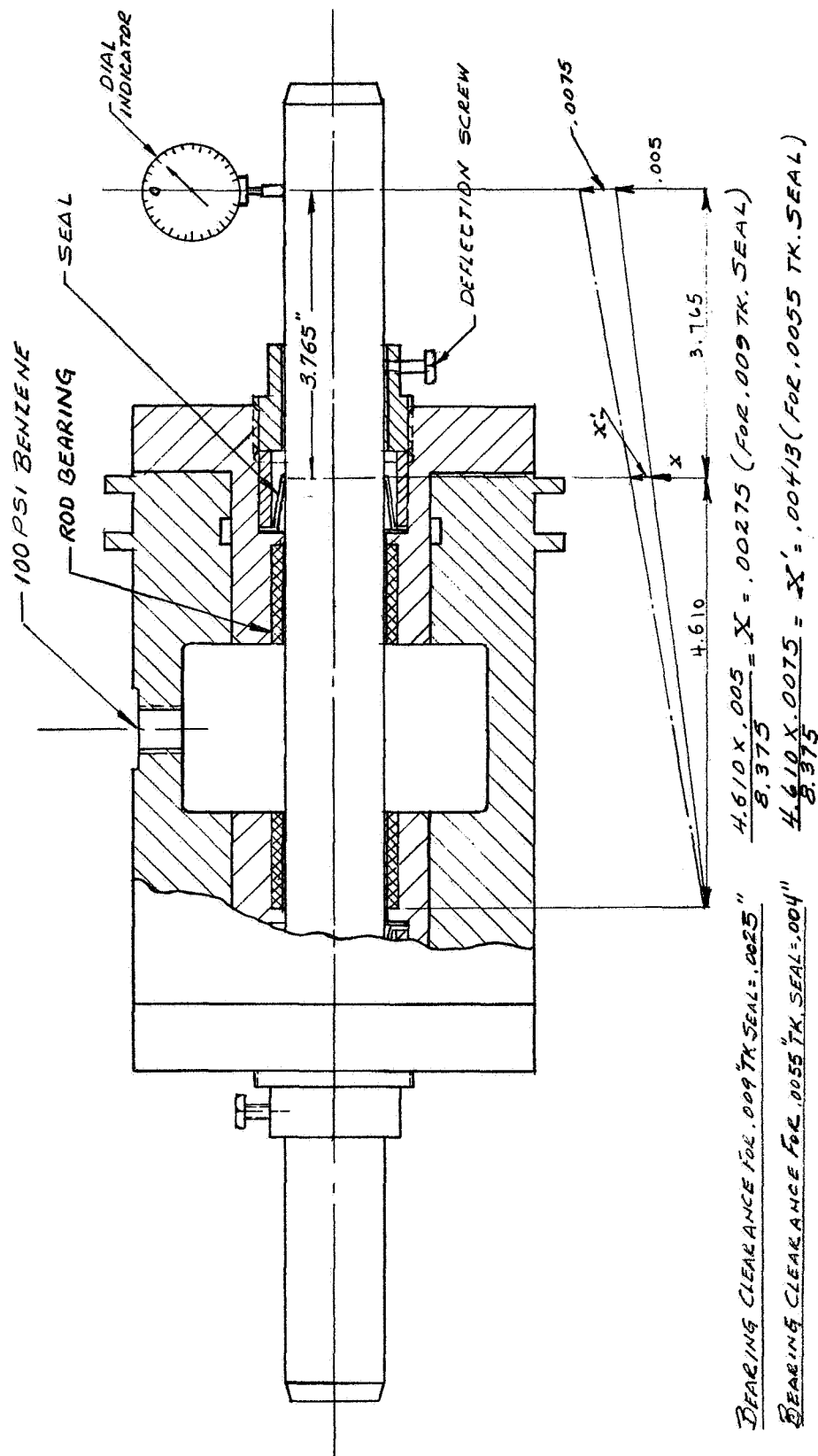


Figure 35. Metallic Seal - Radial Deflection versus Leakage Test Set-up

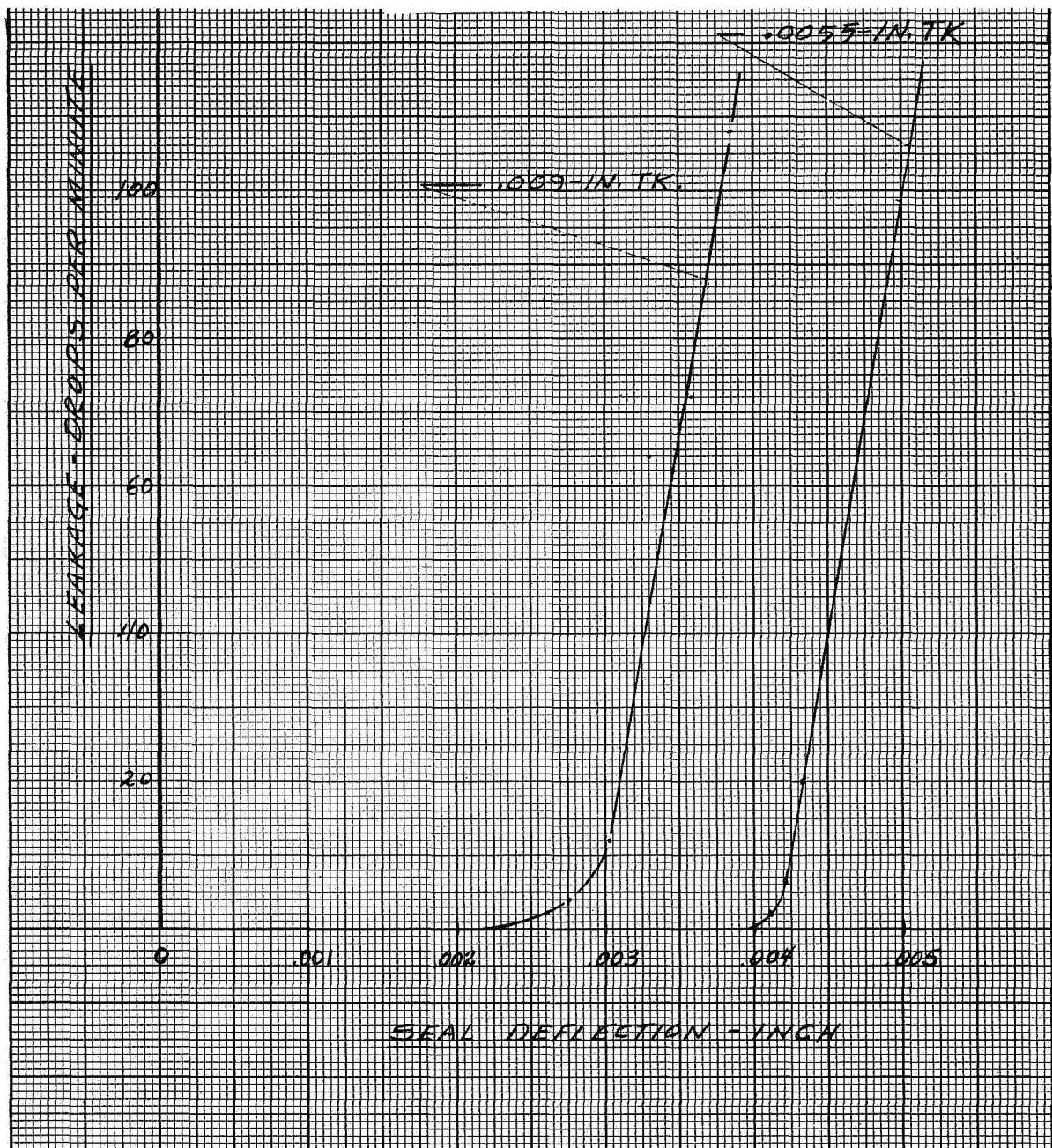


Figure 36. Design D Vascojet 1000
Leakage versus Seal Deflection

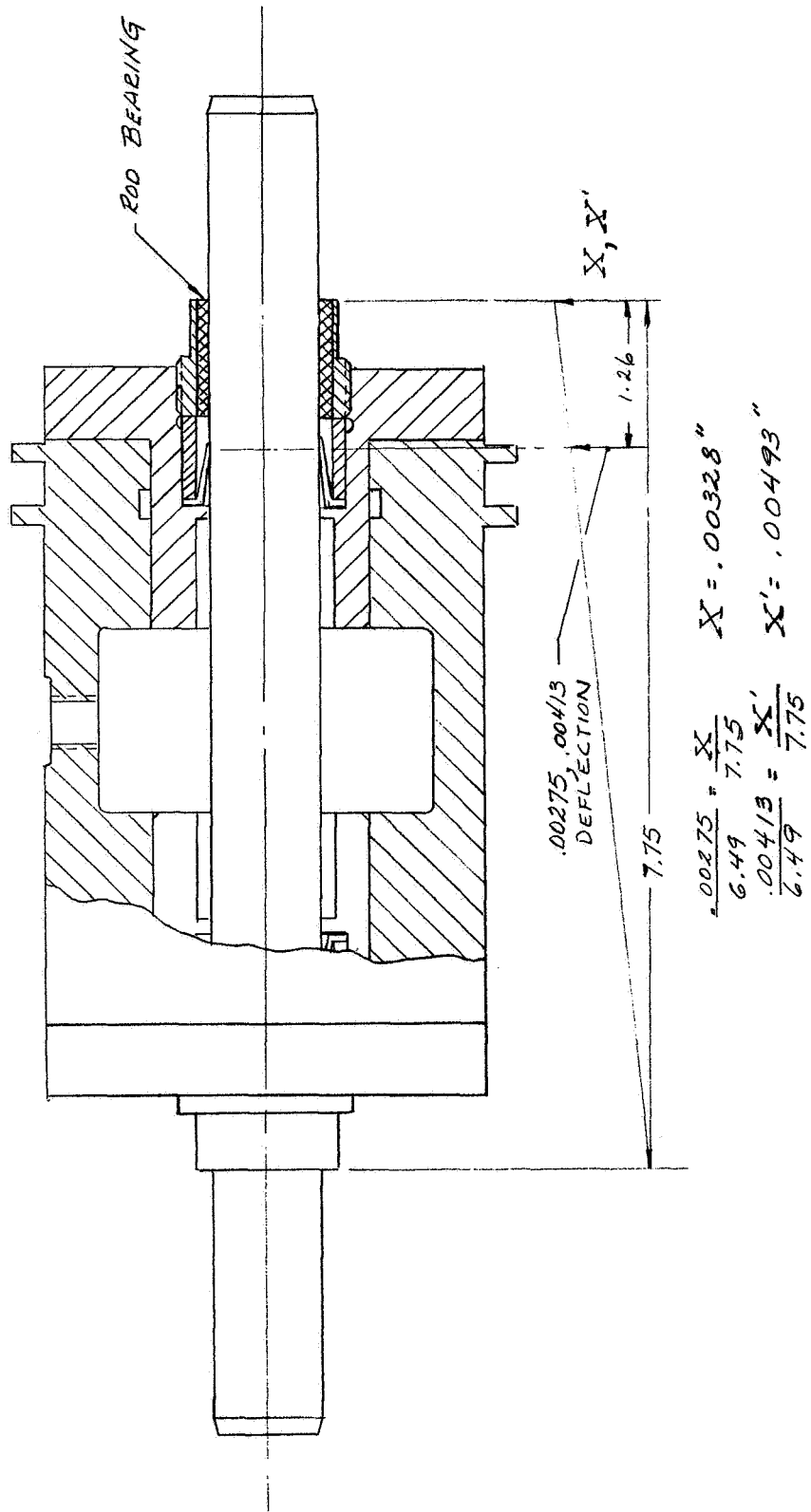


Figure 37. Bearing Outboard of Seal

Modifications to seal design D (metal lip seal) to enable it to withstand additional side motion are also being investigated. Figure 38 depicts a promising approach. This design, which is similar to a ball joint, incorporates a flange nestled in a mating seat. The seat also acts as a static seal. A spring washer provides the seating force at zero pressure conditions. Nickel Foametal impregnated with CaF_2 and BaF_2 was selected as the seat material because of the potential lubricating ability of the impregnant and good conformability of the relatively soft nickel matrix. Figure 39 depicts the position of the seal under exaggerated side motion. Another configuration based on the same approach is shown in Figure 40. However, this design is somewhat complex because of the separate static and dynamic seals required.

As recommended by NASA, seal design D is also being reviewed to determine what possible means can be incorporated into the seal configuration to provide a fail-safe backup for high pressure conditions. Since the seal was designed for optimum low pressure performance, its ability to withstand high pressures is limited. Under normal operating conditions the seal would not be subjected to high pressures. Even with a catastrophic failure of the first-stage high pressure seal, pressure would build up against the second-stage seal only under conditions of high inflow to the actuator that reached the flow limits of the downstream lines.

However, it is recognized that in the design of certain high performance actuators, it may be desirable to build in safety features so that a catastrophic failure of the first-stage seal does not render the component inoperative. Such a feature generally consists of a restrictor-check valve in the return port between the first- and second-stage seal. (The check valve prevents reverse flow in the event of second-stage seal failure.) The restrictor limits the internal leakage flow resulting from first-stage seal failure so that pressure can be maintained in the actuator. Under these conditions the second-stage seal will be required to withstand practically full system pressure.

The critical part of the present lip seal design is at the contact area. This portion of the seal is stressed to approximately 110,000 psi (hoop stress) due to a radial stretch of .0019 inch. (interference fit) over the piston rod. An increase in fluid pressure would increase the induced hoop stress at this point. To prevent

- NICKEL FOAM METAL SEAT

- SPRING

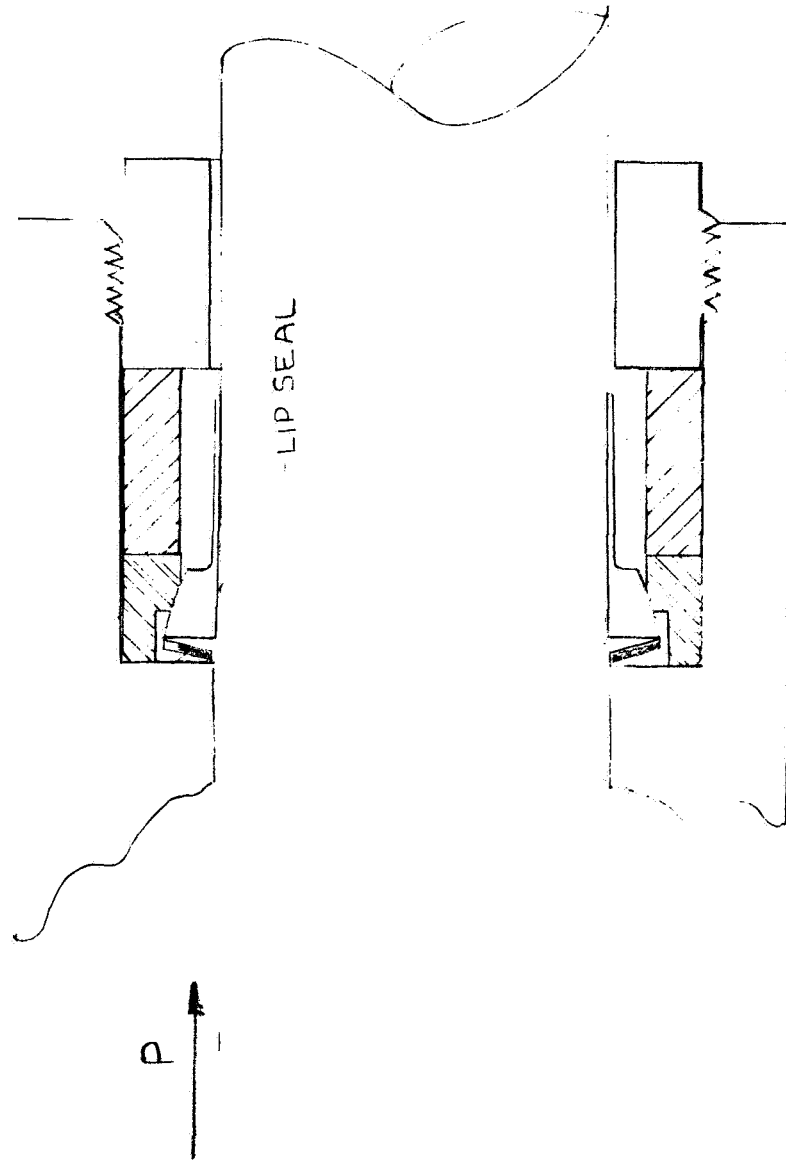


Figure 38. Modified Metal Lip Seal

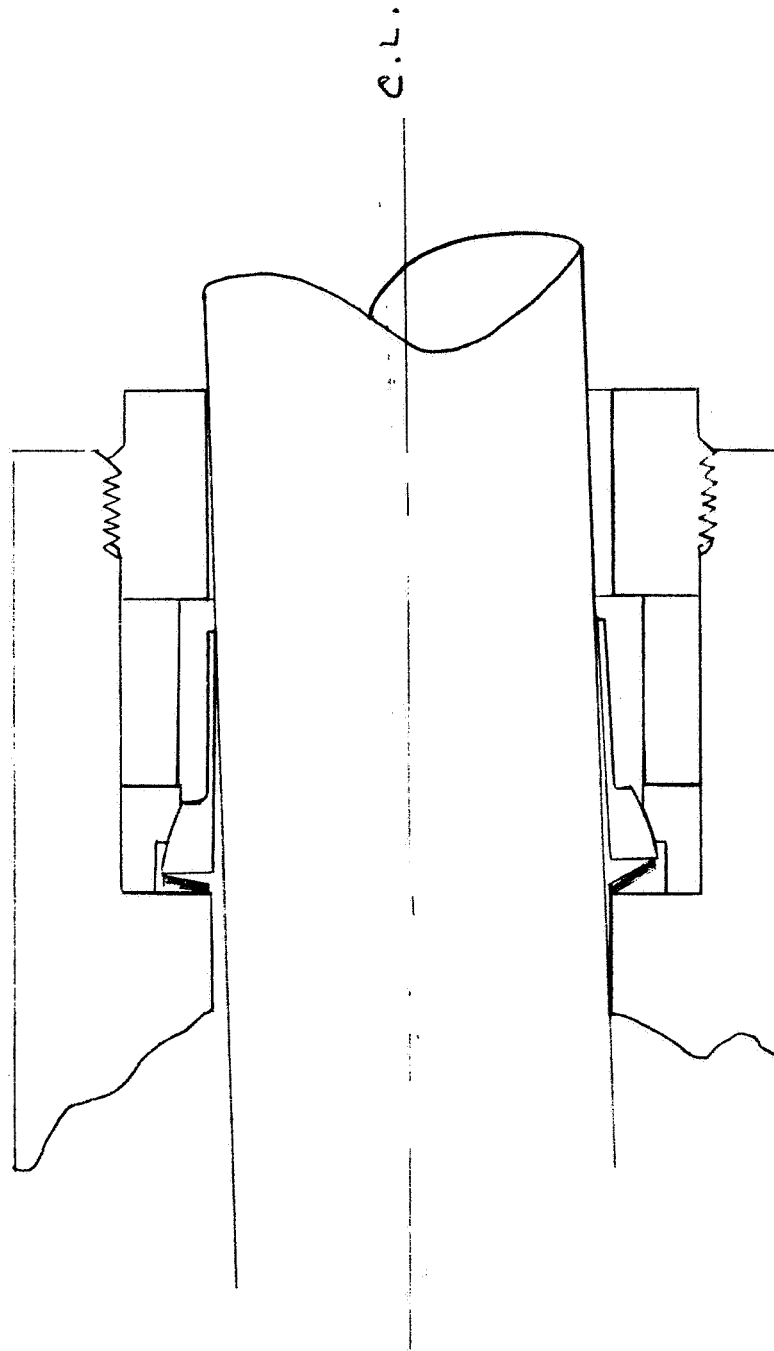


Figure 39. Modified Metal Lip Seal With Side Load

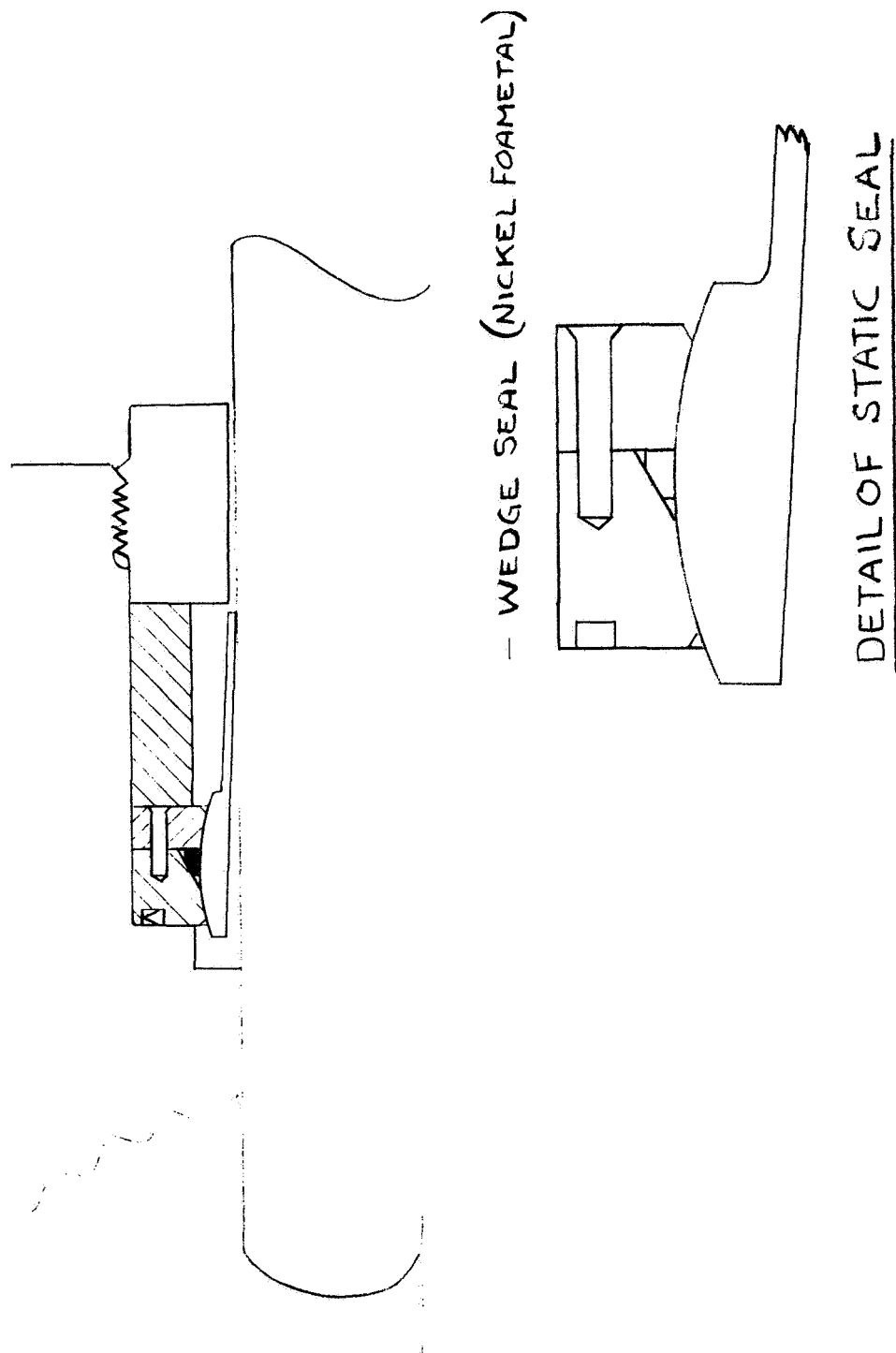


Figure 40. Modified Metal Lip Seal

the seal from being over-stressed a backup device would be required to limit the deflection due to pressure. However, in order not to limit the built-in flexibility of the seal, the clearance between the backup and the seal must be closely controlled.

Based on the assumption that the seal can withstand a maximum hoop stress of 160,000 psi, (tensile yield strength of Vascojet 1000 at 600°F = 183,000 psi) then the allowable increase in hoop stress due to fluid pressure is:

$$160,000 - 110,000 = 50,000 \text{ psi}$$

The fluid pressure required to produce a hoop stress of 50,000 psi is:

$$P = \frac{St}{R} = \frac{50,000 \times .0055}{.5} = 550 \text{ psi}$$

Radial deflection of the sealing lip (from the assembled position) due to a fluid pressure of 550 psi is:

$$R. D. = \frac{R}{E} (S_2 - \nu S_1) *$$

where R = mean radius of seal

E = modulus elasticity (for Vascojet 1000 $E = 23.9 \times 10^6$ at 600°F)

S_2 = hoop stress ($S_2 = \frac{PR}{t}$)

S_1 = meridional membrane stress ($S_1 = \frac{PR}{2t}$)

ν = Poissons ratio = .26

$$R. D. = \frac{.5}{23.9 \times 10^6} (50,000 - (.26 \times 25,000))$$

$$R. D. = .000908 \text{ inch}$$

Based on this analysis, the sealing lip can expand .0009 inch radially without being overstressed. Beyond this point the seal will have to be backed up. In

* "Formulas For Stress and Strain," First Edition, R.J. Roark, p 227

order not to inhibit the seal during low pressure operation or in its ability to follow radial motion, the equivalent clearance will have to be maintained between the seal and backup. Such a clearance would be extremely difficult to machine and maintain upon assembly.

An alternate approach to protecting the low-pressure second-stage seal is to provide a fail-safe backup to the first stage seal. The backup arrangement for the first-stage seal is shown in Figure 41. This configuration incorporates two contracting sealing rings with a vent between them. During normal operation the upstream seal performs the sealing function. Leakage past this seal is returned to the system return line through the vent. The downstream sealing ring provides only nominal sealing during normal operation. Wear on this ring will be minimal since it is not energized. The vent is a calibrated orifice which will accept normal seal leakage without pressure build-up. In the event that failure of the first-stage seal occurs, the increased leakage will saturate the orifice causing pressure to build up and energize the second sealing ring. Thus pressure is maintained in the cylinder for adequate operation.

The above arrangement provides a fail-safe backup without placing tight performance restrictions on the low pressure seal.

F. FIRST-STAGE HIGH PRESSURE SEALS

Several designs of contracting sealing rings are being investigated for possible use as first-stage rod seals in the Task V test phase. These include sealing rings fabricated of Graphitar grade 80, Type 440C stainless steel, Polymer SP, and an alloyed cast iron. Typical designs are shown in Figures 42 through 45.

Static leakage checks were obtained on the sealing rings shown in Figures 42, 43 and 44. These were checked in the one-inch rod size, at pressures between 0 to 4000 psi. Leakage measurements were made with the rings assembled in a single-ended actuator. Leakage characteristics of these rings are shown in Figure 46. The three-piece graphite rings exhibited fairly high leakage at low pressures, but leakage decreased with increasing pressure. The C. Lee Cook

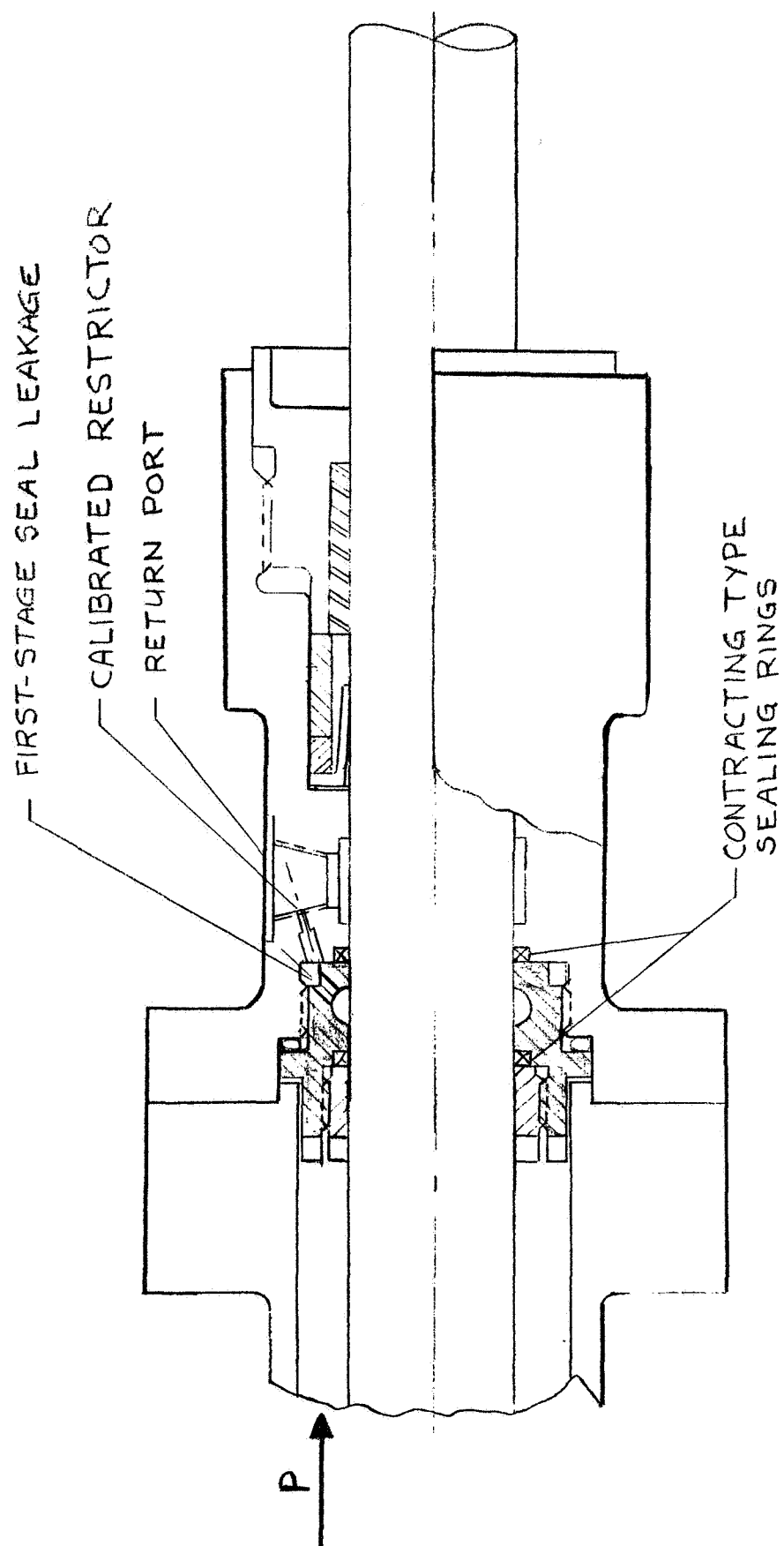


Figure 41. Fail-Safe First-Stage Seal

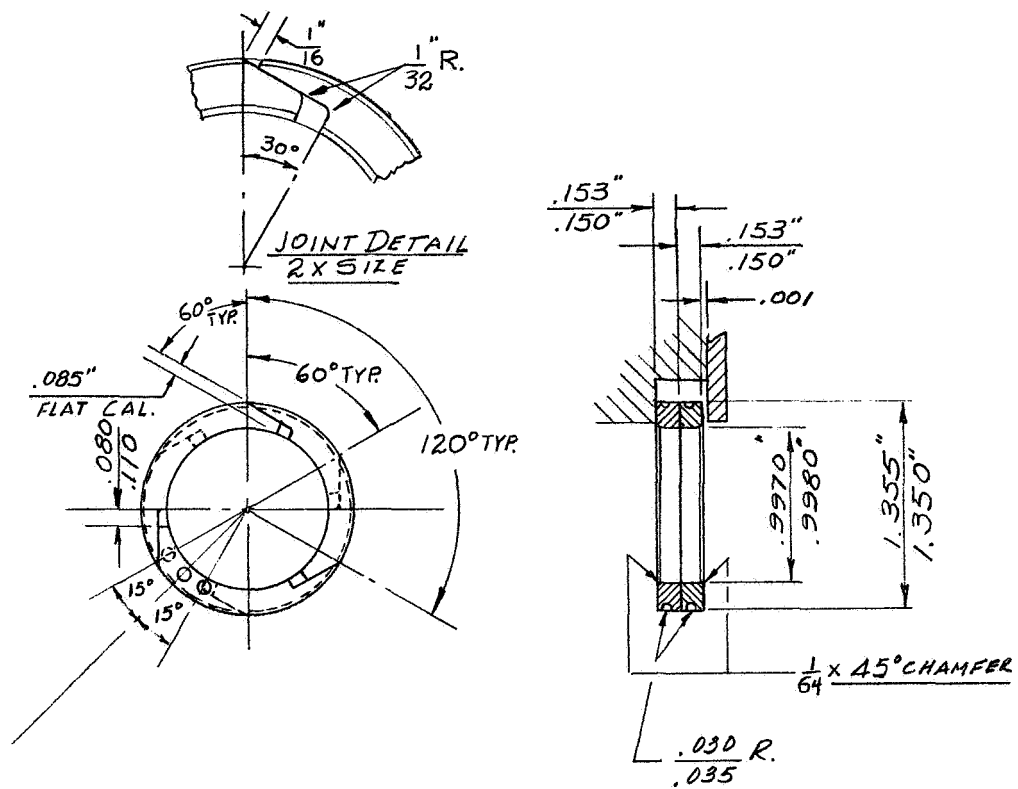


Figure 42. Segmented Graphitar* Seal Ring

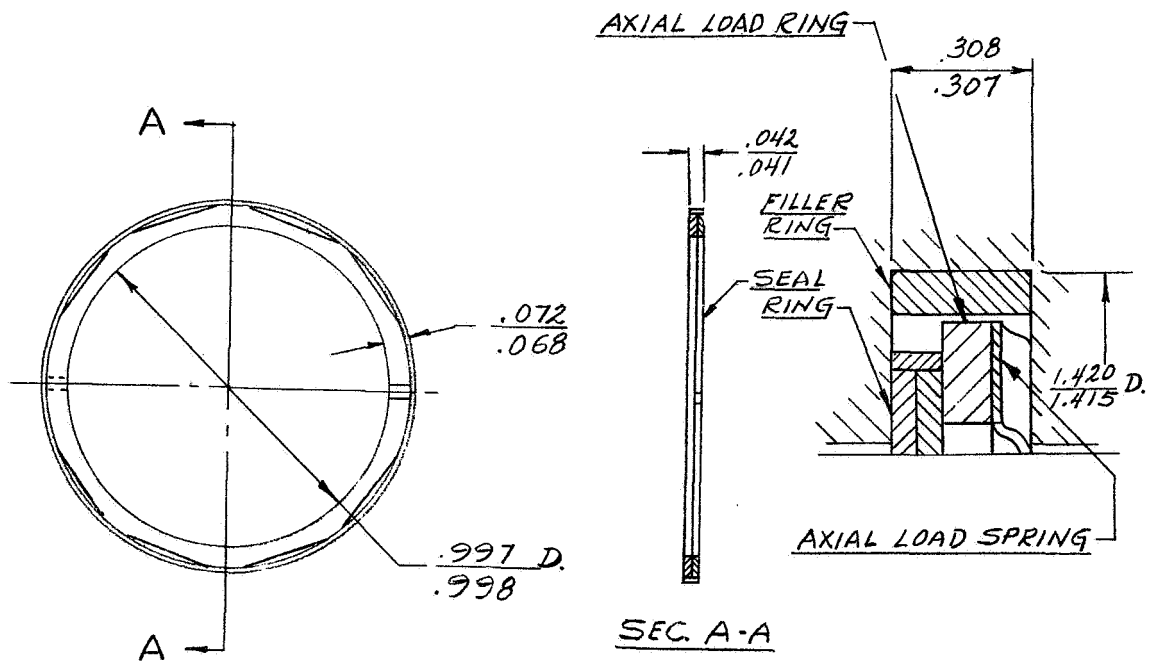


Figure 43. Cook Seal Type MX-1 (Type 440C)

* United States Graphite Co.

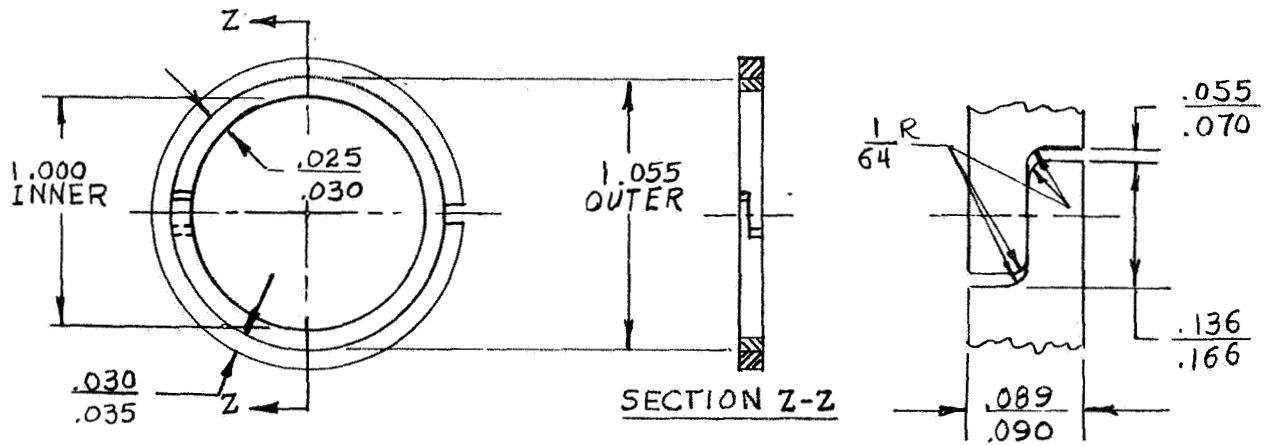


Figure 44. Polymer SP Sealing Ring

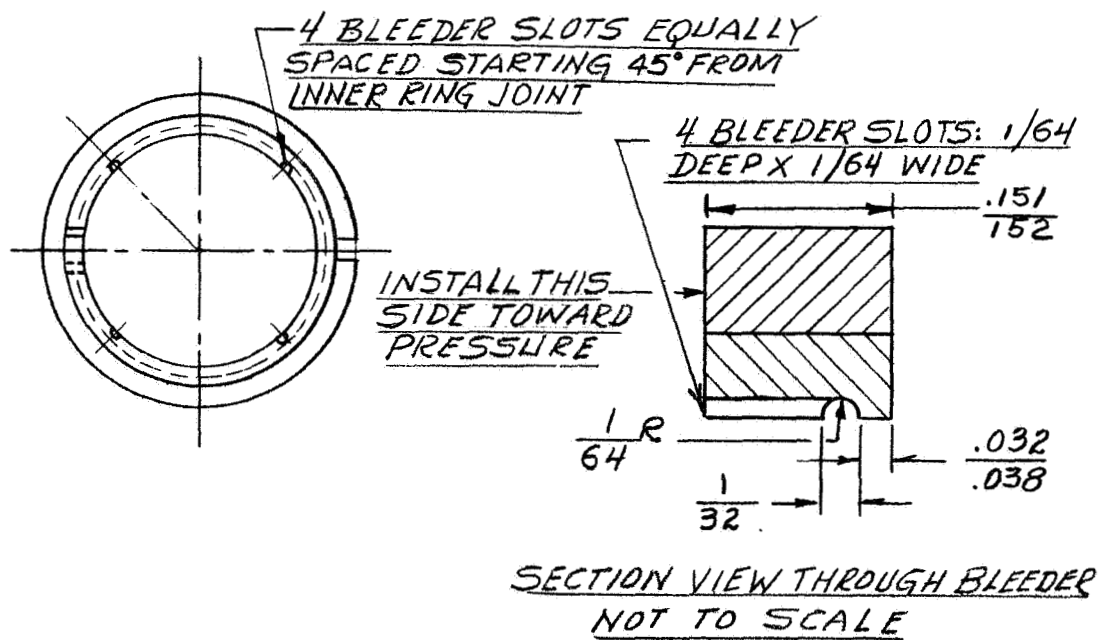


Figure 45. Pressure Balanced Sealing Ring (for 3-inch Rod)

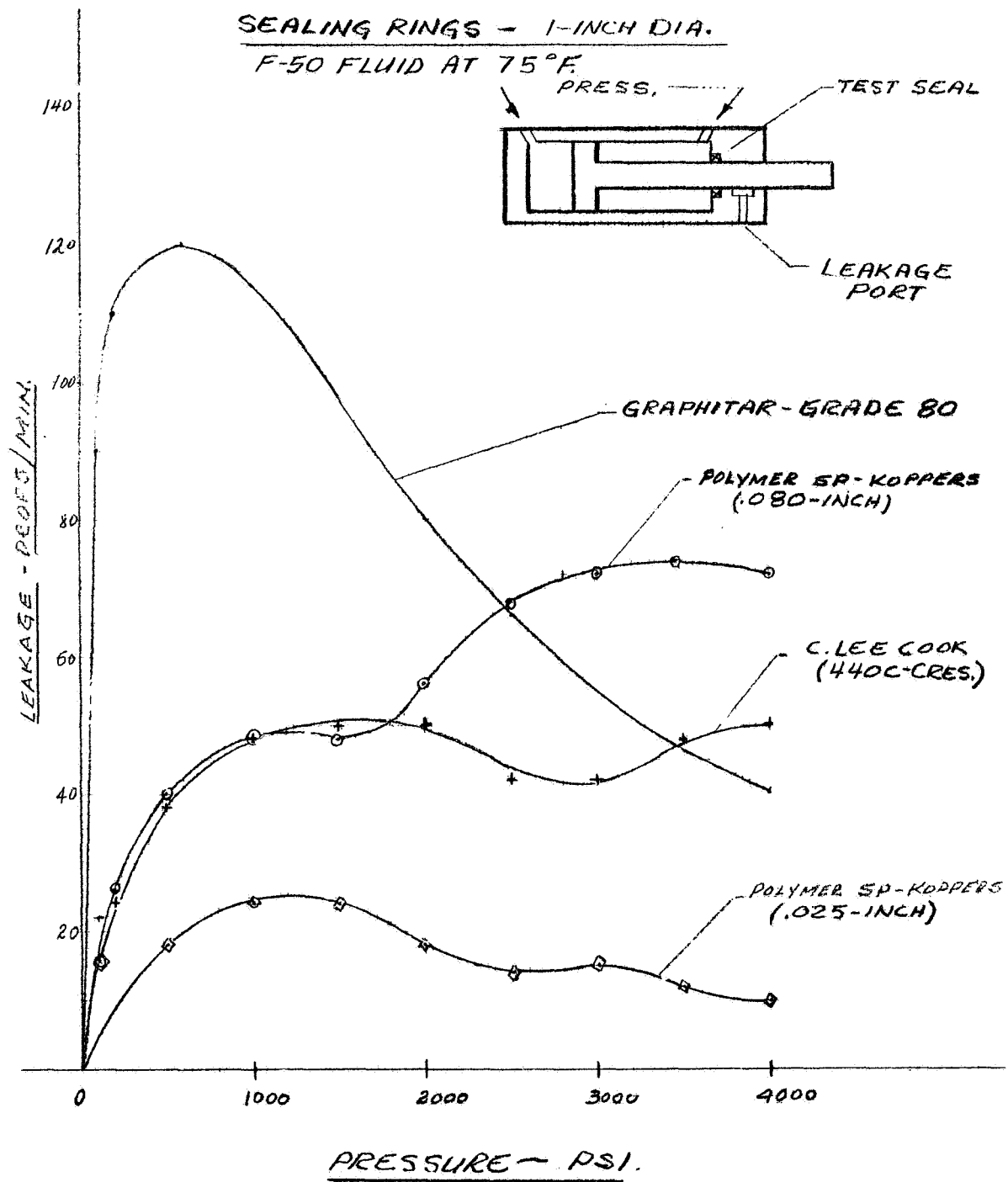
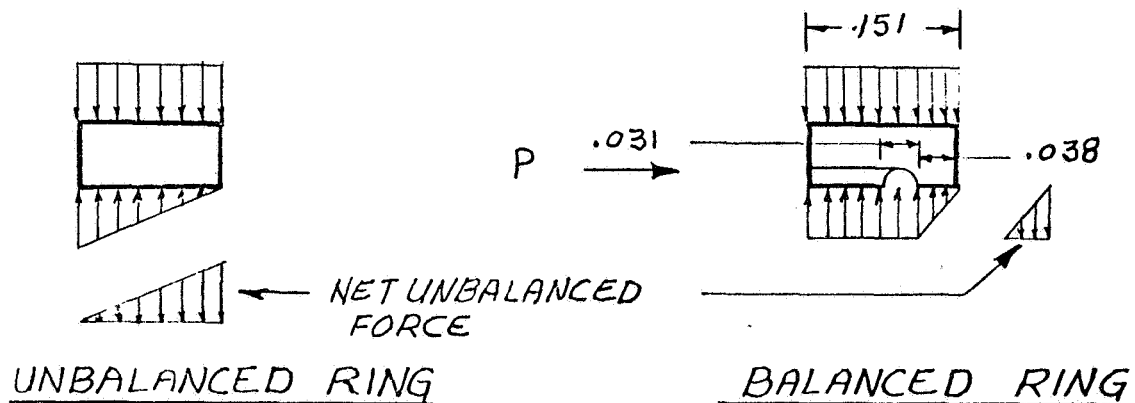


Figure 46. Seal Leakage Test - Contracting Sealing Rings

three-piece metallic sealing exhibited fairly low leakage throughout the pressure range. The two-piece Polymer SP sealing rings (Figure 44) were evaluated in two thicknesses. Results showed that a ring of thinner section provides better conformability and smaller gap area, and consequently lower leakage. Seal friction was not obtainable on these rings due to the unbalanced condition of the single-end actuator used in these tests. Leakage rate and friction of new and used Polymer SP sealing rings of the same configuration were compared. The used rings were tested in a previous high temperature seal program at 500°F fluid (Ref. ML-TDR-64-266). Total test time on these rings was 166 hours, with approximately 120 hours at 500°F. The results shown in Figure 47 were obtained with the sealing rings assembled in the double-ended actuator used in the above referenced program. As shown in Figure 47, leakage rates of the used rings were slightly higher at 2000 psi. At 3000 psi the used rings exhibited lower leakage than the new rings. Seal breakaway friction varied slightly for the new and used rings at pressures to 2000 psi.

In view of the 3000-hour life requirements for these rings, it is believed that pressure balancing will be desirable to minimize ring wear. Figure 45 depicts a pressure balanced version of the Polymer SP sealing ring proposed by Koppers Company Inc. for the three-inch rod size. This design provides approximately 67 percent pressure balancing. The difference in contact pressures between a balanced and unbalanced ring of this configuration is depicted below.



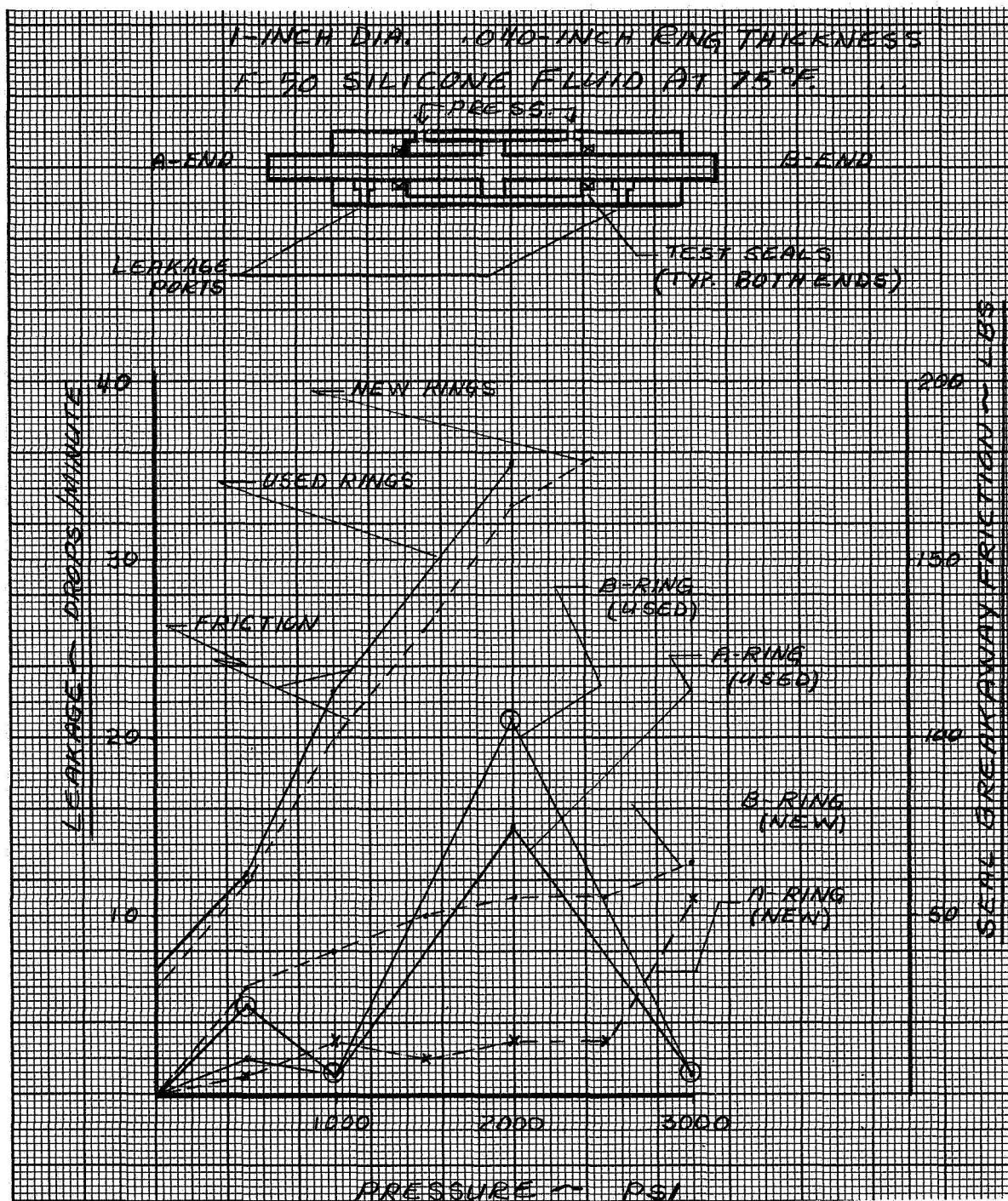


Figure 47. Leakage and Friction of New and Used Polymer SP Contracting Sealing Rings

Assuming the pressure gradient to be linear, the contact pressure for the unbalance ring is:

$$P_c = \frac{4000}{2} = 2000 \text{ psi}$$

For the balanced ring, the contact pressure is reduced to:

$$P_c = \frac{\frac{4000}{2} \times .038}{.151 - .038} = 666 \text{ psi}$$

In the above calculations the contact pressure generated by the outer ring (shown in Figure 45) was not included, since the spring load (approximately four pounds) was negligible. However, it can be seen that the bearing pressure is greatly reduced by pressure balancing. With a pressure balanced ring, the thickness of the ring is adjusted to keep the axial forces in balance with the radial forces. This ensures adequate closing of the ring in the radial direction and at the same time maintains ring contact with the gland wall.

A metallic (alloyed cast iron) version of this design is also being considered. Another design being considered, Figure 48, is also pressure balanced, incorporates garter springs for radial loading and a friction backer ring for axial springing.

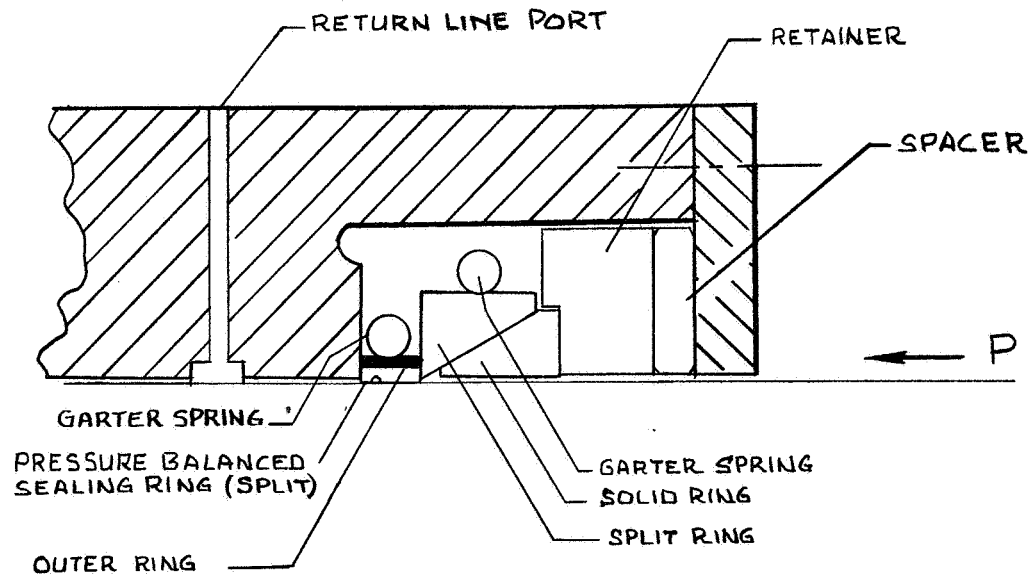


Figure 48. Pressure Balanced Sealing Ring with Axial Loading Rings

The split sealing ring is pressure balanced to reduce friction and wear. The friction backer ring, which consists of one split wedge ring and one solid wedge ring, provides the axial springing to the seal to maintain contact with the groove side. The backer ring assembly is pressure balanced. Testing of this configuration with Oronite fluid by the Koppers Company indicated leakage of approximately 30 cc/hr. from 300 to 3000 psi at 450°F.

TASK IV - LOW PRESSURE SEAL TESTING

A. GENERAL

As mentioned in the discussion of Task I, operational checkout of the one-inch seal test rig was accomplished at room temperature, 400°F, 500°F, and 600°F. The strain gauge for measuring seal friction has been calibrated at room temperature and is being installed in the test rig for checkout at elevated temperatures.

B. TEST PARAMETERS

Test parameters for the low pressure evaluation of candidate seal designs have been formulated. The test profile is shown in Figure 49. Testing will be conducted with F-50 silicone fluid at 100 psi and at temperatures of 400°F, 500°F, and 600°F. Test time for each of these temperature levels will be 50 hours or until failure. Seal leakage in excess of one drop per minute or a two fold increase in friction is considered failure.

As shown in Figure 49, the test profile consists of the following operation:

- 1) Long-stroke cycling \pm two inches at 30 cpm during the heatup from room temperature to the designated temperature level.
- 2) Continuous operation at the designated temperature level for 50 hours. Operation consists of alternate short-stroke cycling (± 0.1 -inch at 300 cpm for two hours) and long-stroke cycling (± 2 inches at 30 cpm for one hour).
- 3) Long-stroke cycling during the cool-down from the designated temperature level to room temperature.
- 4) Repeat steps 1, 2 and 3 for the next temperature level.

Seal leakage and friction will be monitored during the operation.

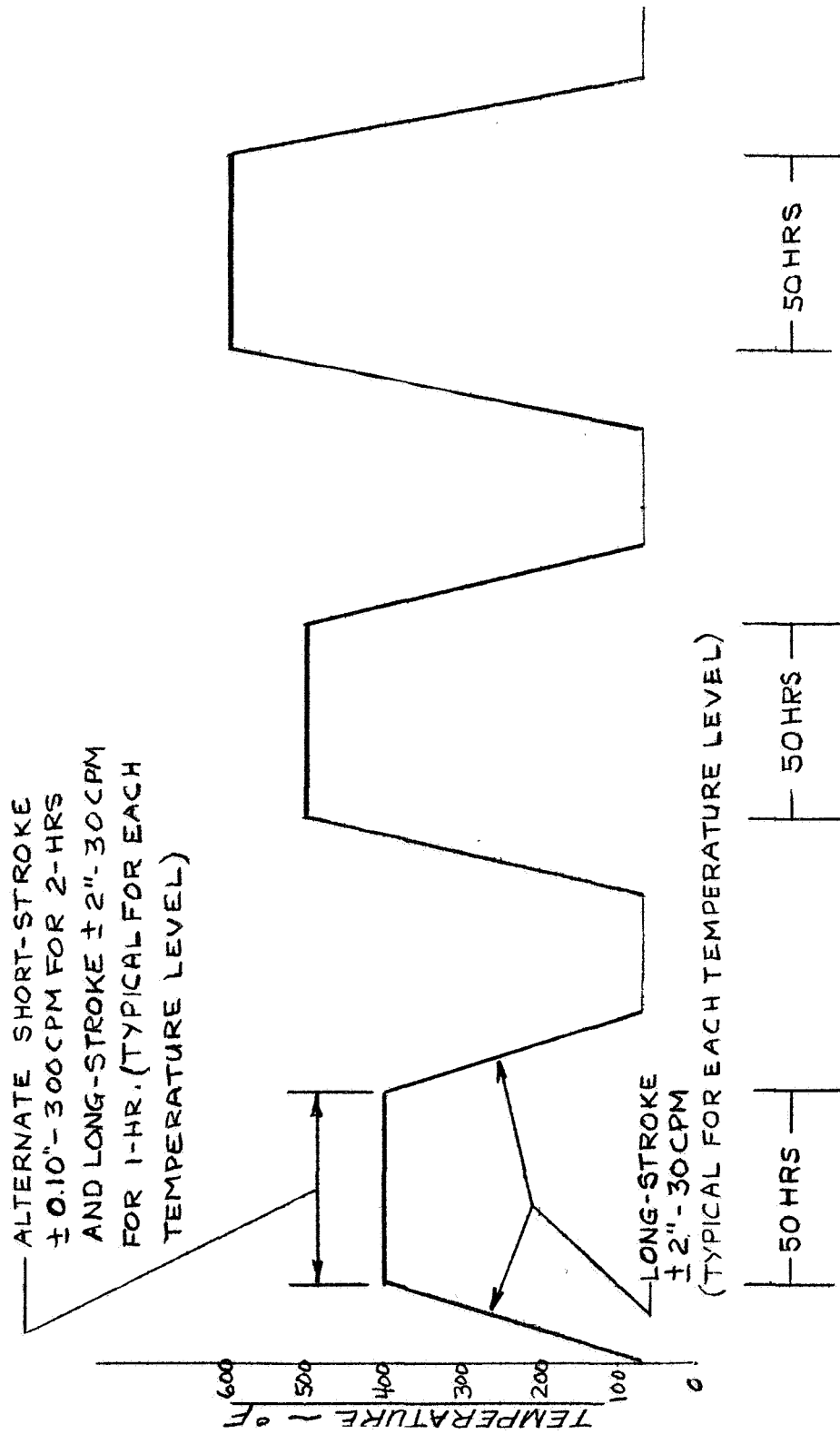


Figure 49. Test Profile - Low Pressure Seal Test

FUTURE EFFORT

The schedule for the next six-month period of the program is shown in Figure 50 .

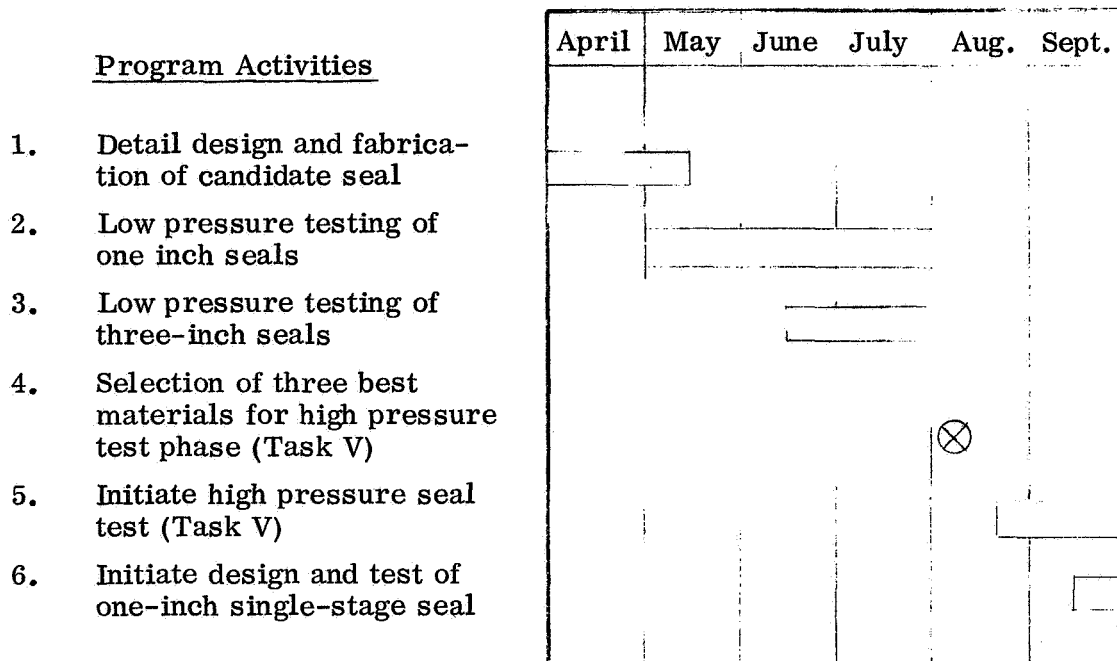


Figure 50. Schedule of Activities for Next Six-Month Period

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5. Republic Aviation Corporation, "Investigation of Techniques for 1000°F Hydraulic Systems," ASD-TDR 62-674, 1962.

APPENDIX A

EVALUATION OF SECOND-STAGE SEAL DESIGNS

A. INTRODUCTION

The basic criteria, method, and results of the evaluation of second-stage seal designs are presented in the following sections. The designs that were considered represent a spectrum of approaches. These approaches were found to fall into four basic configuration categories: wedges, lips, reeds, and specially shaped elements such as a V, C or X.

The state of the art of high temperature rod seal design is still primarily empirical. Until recently, little has been done to generate rational methods of analyses for the many conceivable combinations of shapes and materials. In addition, published data are limited largely to test results in terms of friction, leakage, and life. Therefore, the analysis of seal designs relies heavily on experience, simple approximations in calculations, and intuition.

An effort was made to optimize the various approaches, considering the materials available as well as the specific application in mind. However, the number of combinations and possible variations precluded extensive study of any one design. Rather, efforts were concentrated on the designs that reached the greatest degree of development and consequently offer the greatest potential for meeting the goals of this program. By applying various practical criteria, a manageable number of likely candidates have been selected that still cover a broad variety of approaches.

B. BASIC APPROACH

A positive-contact type of second-stage rod seal is required if leakage is to be kept to very small values (one drop per minute at failure). Experience shows that this goal is quite difficult to attain, particularly in the larger rod

sizes. Furthermore, it must be met over a broad range of pressures, from 0 (during system shutdown) through nominal operating (return line) pressure, to surges of several times nominal pressure; and over a fluid temperature range of -40°F to $+600^{\circ}\text{F}$.

An overall seal configuration is selected primarily to attain the desired loading of the sealing surface at the rod surface, and to accomplish other functions such as static sealing and alignment. The seal material's mechanical properties govern its ability to transmit an adequate sealing stress to the rod over the entire pressure and temperature ranges, and to continue to deliver this sealing stress after a reasonable amount of wear occurs. The seal's wear rate will then determine its useful life. The configuration must also overcome practical limitations such as adverse tolerance buildups, eccentricities and out-of-roundness of the rod. These practical problems in effect reduce the seal's theoretical ability to accommodate wear.

Allowable stress levels that are appropriate to the particular seal material provide certain design constraints on the configuration. These stresses will generally fall into one of the following categories:

- 1) Hoop stresses that limit the radial deflection available for wear compensation.
- 2) Combined stresses that limit the transfer of loading energy to the sealing surface.
- 3) Sealing (bearing) stresses that produce acceptable friction and wear rates.

The principal tradeoffs to be considered in selecting seals are described below:

1. Wear Rate Versus Wear Compensation. The need for the sealing surface to deflect radially is proportional to its wear rate. A fairly high wear rate, coupled with a considerable ability to deflect, might be preferable to a low wear rate and little ability to deflect. Compensation for wear is obtained if the material is elastic and if the loading method does not induce excessive local stresses in some other part of the seal element.

2. Leakage Versus Friction and Wear. Bearing stresses at the seal-rod interface that will produce a low-leakage seal will vary with different materials. The configuration must attain a satisfactory stress level throughout wide ranges of temperature and pressure without causing excessive friction and wear. Any seal design is a compromise among many possible values of stress that can be experienced over the attainable environmental range, and both before and after a change in cross-section due to wear occurs.

3. Reliability Versus Performance. Redundant elements, conservative allowable stresses, and special provisions to support the seal during pressure surges are desirable features from the standpoint of reliability. The effect of such features must be balanced against any adverse effect they may have on friction, wear or compensating ability. Furthermore, care must be taken that an inherent weakness that can result in abrupt failure is not built into the design. Any penalty resulting from a necessary compromise in design will be made in such a manner so as to preclude any built in catastrophic failure potentials. However, by compromising one feature to improve another, it is possible that disadvantages such as reduced service life, or increased allowable leakage or friction may occur.

4. Performance at Nominal Versus Design Conditions. The seal's specific duty cycle may dictate additional compromises in its design. For example, in flight control actuators, friction is frequently a dominant factor. This may lead to a deliberate reduction in initial loading and, consequently, a foreshortened life due to the early occurrence of high leakage rates. Utility actuators can tolerate higher friction (assuming this does not result in excessive wear), but usually experience a relatively low number of cycles during the vehicle life. The seals for these two applications may thus differ considerably from one another in design.

In summary, the seal configuration must exploit the material's strong points while offsetting its weak points to an acceptable degree. Any gross mismatch among the various design factors will render the seal ineffective.

C. SEAL RATING SYSTEM

The foregoing considerations were used as the basis for setting up the seal rating system. This system assists in making a preliminary selection of designs

that warrant further development effort. Candidates are rated on certain factors deemed essential to attaining program objectives. Several candidates are thereby eliminated from further consideration because of an unacceptable quality or because of low overall score. The remaining designs are then rated on the basis of other factors, considered desirable but not absolutely essential, to aid in the final selection.

This system does not assure the identification of a distinct order of merit among the various candidates. Assigned values for many of the factors are estimated and, consequently, are subject to argument. However, the rating does identify the least promising candidates and reduces the final group to a manageable size. Final selection must to some extent depend on intuition for the most practical approaches.

Scoring of factors is limited to a three-digit spread (0-1-2). The lack of quantitative data precludes precise scoring; thus, more refined gradations are considered impractical at this time. The standard of grading is relative to the group average. The evaluation covers all elements of the installation (such as static and dynamic seals, bearings and loading devices) that affect actuator design.

D. ESSENTIAL FACTORS

1. Sealing Mechanism. The seal conforms to the rod by means of a feasible loading technique. Appropriate stress levels and patterns are generated in the seal element throughout the temperature and pressure range. Geometry and behavior of elements can be predicted and controlled within workable limits. Effective static sealing is available. Pressure does not increase leakage.

2. Wear and Compensation. The seal exhibits either a low wear rate or the ability to compensate for wear, and wears rod surface at a low rate. The wear process is not detrimental to overall performance of seal installation (i.e. no adverse effect on static sealing). Pressure does not increase wear.

3. Reliability. Seal failure is slow and detectable. Seal construction is rugged and inherently resistant to abuse in use. Redundant sealing elements available.

Precise definition of operating conditions and behavior is not critical; possesses inherent latitude to withstand pressure surges.

4. Short-term Potential. Knowledge of concept is available or can be acquired in the near future. Configuration can be optimized and evaluated within time period of program.

E. DESIRABLE FACTORS

1. Design Features. Materials exhibit compatible coefficients of expansion. Rod does not require exotic plating. Friction is relatively low (for good servo performance). The seal has a relatively high degree of self-compensation for wear. Geometry lends to pressure balancing or pressure relief.

2. Cost. Relatively easy to manufacture (reasonable tolerance requirements, accessible geometry for machining). Materials are available and machineable.

3. Serviceability. Easy to install and remove. Does not require custom tailoring; no special installation tools required.

4. Past experience with similar approach.

F. SEAL RATINGS

Rating work sheets for each seal design are attached (pages A-24 to A-41). Final overall scores are summarized in Table A-1. In making the final selection, a minimum overall score of 10 was established for acceptance. Five seal configurations and one alternate were selected on this basis. The seal-material combinations recommended for further development are shown in Table A-2. Since certain seal designs were applicable to more than one material, alternate combinations are also included.

The final recommendations represent a broad coverage of the various available seal concepts and materials. It is possible that other seal-material combinations may also offer considerable potential. However, it would not be feasible to evaluate all possible combinations within the scope of the program.

The final selection of configurations was influenced by past and recent experience, and are considered to be representative of the best features of the many designs that were reviewed.

The recommended seal designs and their appropriate materials are provided below.

1. Design B: V-seal/Polymer SP

This design consists of three V-shaped sealing elements, a load ring, back-up ring, and loading springs. The latter provides the loading force to effect a seal and also acts as a wear compensating device. Polymer SP V-seals have undergone substantial development and evaluation in previous seal programs. Results have been quite satisfactory at temperatures up to 600°F. Low friction and low wear characteristics of the SP material offer good potential for long-life operation. Nickel Foametal impregnated with $\text{CaF}_2 + \text{BaF}_2$ is suggested as an alternate material with this design.

2. Design D: Lip Seal/Vascojet 1000

3. Design D: Lip Seal/Cobalt-Molybdenum Alloy

This seal (2 & 3 above) utilizes an interference fit over the rod to effect a seal. The stresses induced by the interference provide compensation for wear. This design also provides a contact width of approximately 0.030 to 0.040 inch on assembly. This enables the sealing force to be distributed over a relatively large area, which results in lower contact stresses. By having the sealing lip facing away from the fluid, excessive buildup of contact stresses due to fluid pressure is avoided. This arrangement also permits relieving of the contact load at the seal interface since fluid pressure will tend to open the inner diameter of the seal. Such a feature is advantageous in reducing friction and wear. Recent test results (Ref. Progress Reports No. 7 and 8 for the present contract) indicate the feasibility of this design.

4. Design F: Spring-Loaded Lip Seal/Silver Alloy Material

This design consists of a truncated-cone. Independent loading of the static and dynamic portions of the seal is accomplished by using separate adjusting

nuts. Finer load adjustments are thus maintained. Spring loading of the sealing lip provides compensation for wear. Preliminary testing of this design using silver alloy as the seal material has indicated the feasibility of this loading arrangement. Relatively light loading was required to effect sealing with a .025-inch thick seal. Seal breakout friction at 0 psi was approximately 40 pounds. Alternate materials recommended for this configuration are nickel Foametal impregnated with $\text{CaF}_2 + \text{BaF}_2$, and Polymer SP.

5. Design I: Wedge Seal/Nickel Foametal Impregnated with $\text{CaF}_2 + \text{BaF}_2$

This design consists of a wedge shaped (45°) sealing element and loading springs. The spring load provides the axial force to effect a static seal at the tapered seat. The radial force component provides the load at the seal-rod interface. The spring washers also provides the means for wear compensation. Testing of this configuration was conducted in previous high temperature seal programs at temperatures of 800°F and 4000 psi and achieved promising results. Alternate material recommended for this design is the silver alloy (72%Ag + 28% Cu).

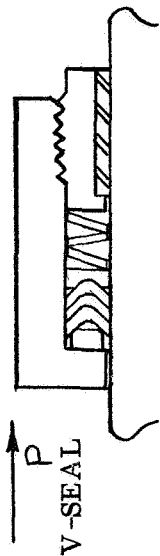
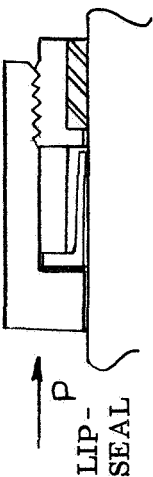
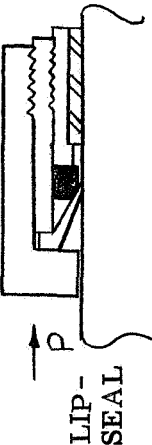
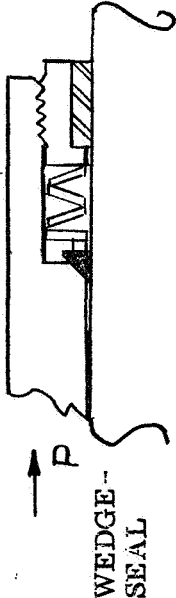
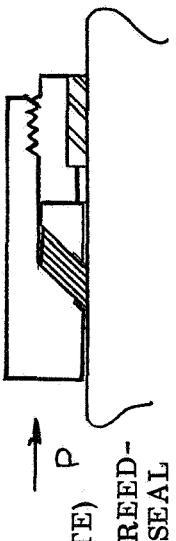
6. (Alternate Design), Design AH: Reed Seal/Vascojet 1000 and Silver Alloy (72%Ag + 28% Cu)

This configuration is recommended as an alternate to be used with a combination hard and soft metal. The seal consists of alternate elements of hard and soft metal. The purpose of the soft metal is to provide added conformability. The sealing elements are approximately .005 to .007 inch thick. Wear compensation is inherent due to the interference fit and the action of fluid pressure. An alternate combination recommended for this design consists of cobalt-molybdenum alloy and silver alloy (providing the cobalt material can be obtained in sheet form). Polymer SP is also recommended. However, the latter would be used in conjunction with Design AH, which is similar to Design H except for the steeper angle, because of the limited ability of the SP sheet material to be formed to a low angle.

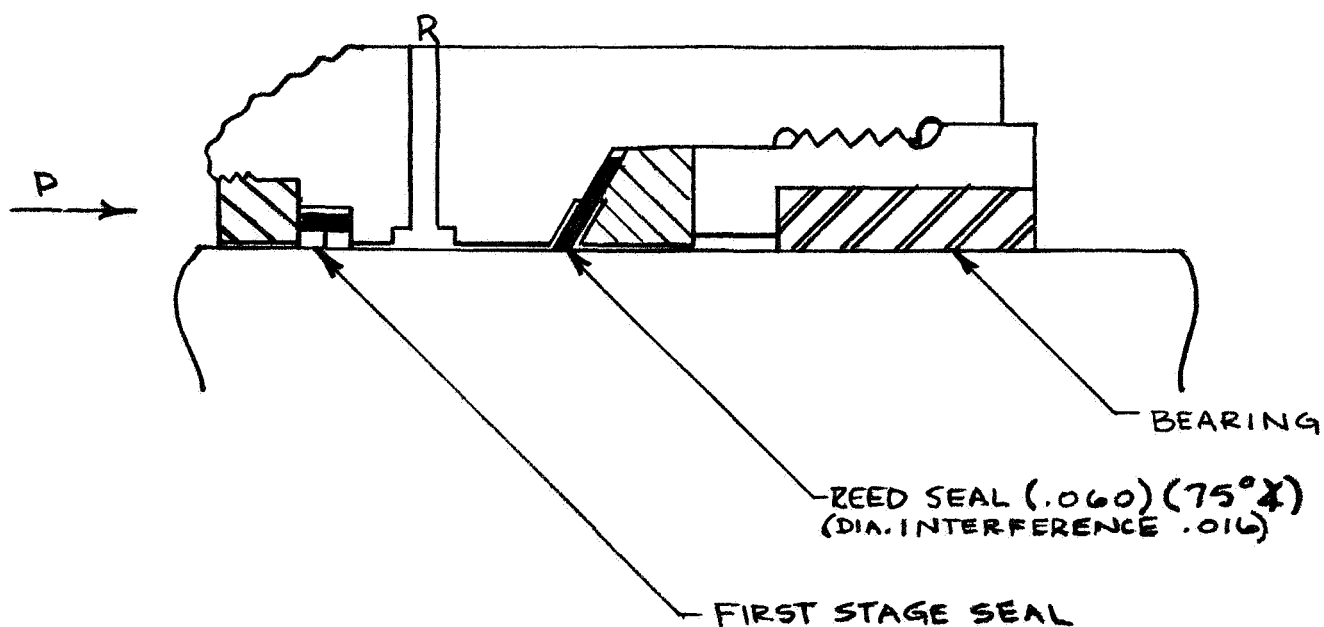
TABLE A-1. SUMMARY OF OVERALL SCORE

Materials		Plastics			Soft Metal							Hard Metal									
Seal Config.	A	B	AH	B	C	F	G	I	IR	L	B	D	E,H	F	J	K	M	N	O		
Essential Factors																					
	1	2	2	2	1	2	1	2	1	2	2	2	2	2	1	1	2	1	1		
	2	2	2	2	1	1	1	2	2	1	1	2	1	1	1	1	2	2			
	3	1	2	2	1	1	1	1	2	1	1	1	2	1	2	1	1	1	1		
	4	2	2	2	2	2	1	1	1	2	2	2	2	1	1	1	1	1	1		
Subtotal	7	8	8	8	5	6	4	6	6	6	6	7	7	5	5	4	5	5	5		
Desirable Factors																					
	1	1	1	1	1	1	-	1	1	1	1	2	1	1	1	-	1	2	2		
	2	2	1	2	1	1	-	1	1	1	1	1	2	1	1	-	0	1	1		
	3	1	1	1	1	1	-	2	2	1	1	0	2	1	1	-	1	1	1		
	4	1	2	1	2	0	1	-	2	2	1	2	2	2	1	1	2	1	1		
Subtotal	5	5	5	5	3	4	-	6	6	4	3	7	6	4	4	-	4	5	5		
Total Score	12	13	13	13	8	10	4	12	12	10	9	14	13	9	9	4	9	10	10		

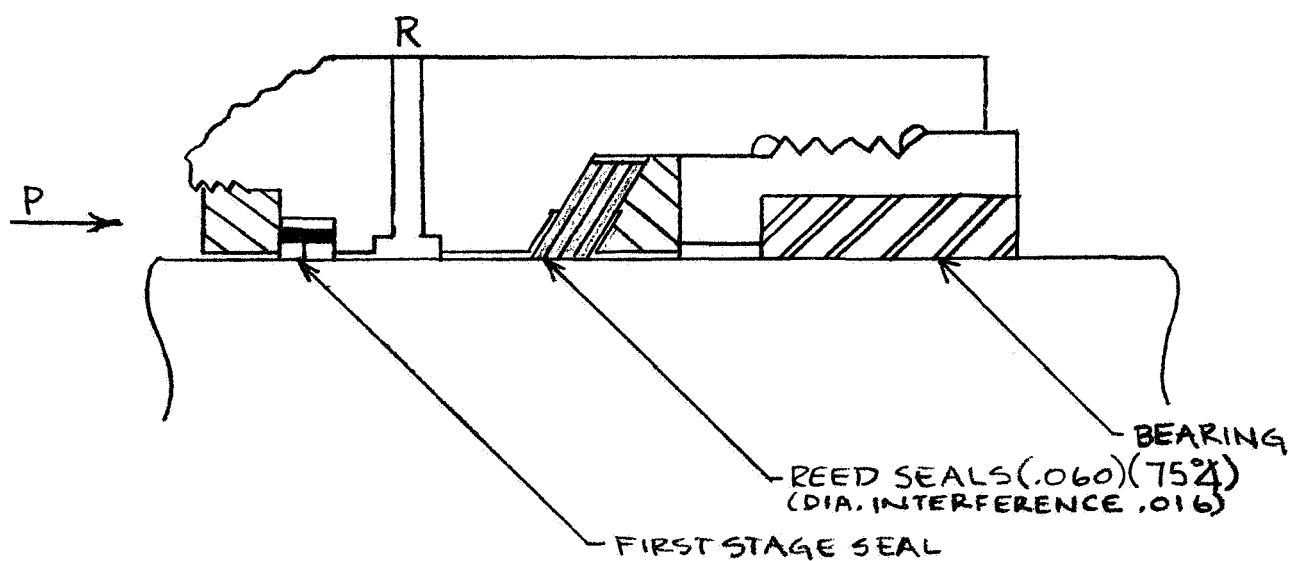
TABLE A-2. RECOMMENDED SEAL DESIGNS AND MATERIALS

SEAL DESIGN	SEAL TYPE	RECOMMENDED MATERIALS	ALTERNATE MATERIALS
B	 V-SEAL	Polymer SP	Nickel Foametal with $\text{Ca F}_2 + \text{Ba F}_2$
D	 LIP-SEAL	Vascojet-1000	
D	SAME	Cobalt-molybdenum alloy (75% Co + 25% Mo)	
F	 LIP-SEAL	Silver alloy (72% Ag + 28% Cu)	Nickel Foametal with $\text{Ca F}_2 + \text{Ba F}_2$ Polymer SP
I	 WEDGE-SEAL	Nickel Foametal with $\text{Ca F}_2 + \text{Ba F}_2$	Silver alloy (72% Ag + 28% Cu)
(ALTERNATE) H or AH	 REED-SEAL	Vascojet - 1000 and silver alloy (72% Ag + 28% Cu) combination	Cobalt-molybdenum alloy and silver alloy (72% Ag + 28% Cu) combination Polymer SP

12-2-65 A



DESIGN A - SINGLE REED SEAL-PLASTIC

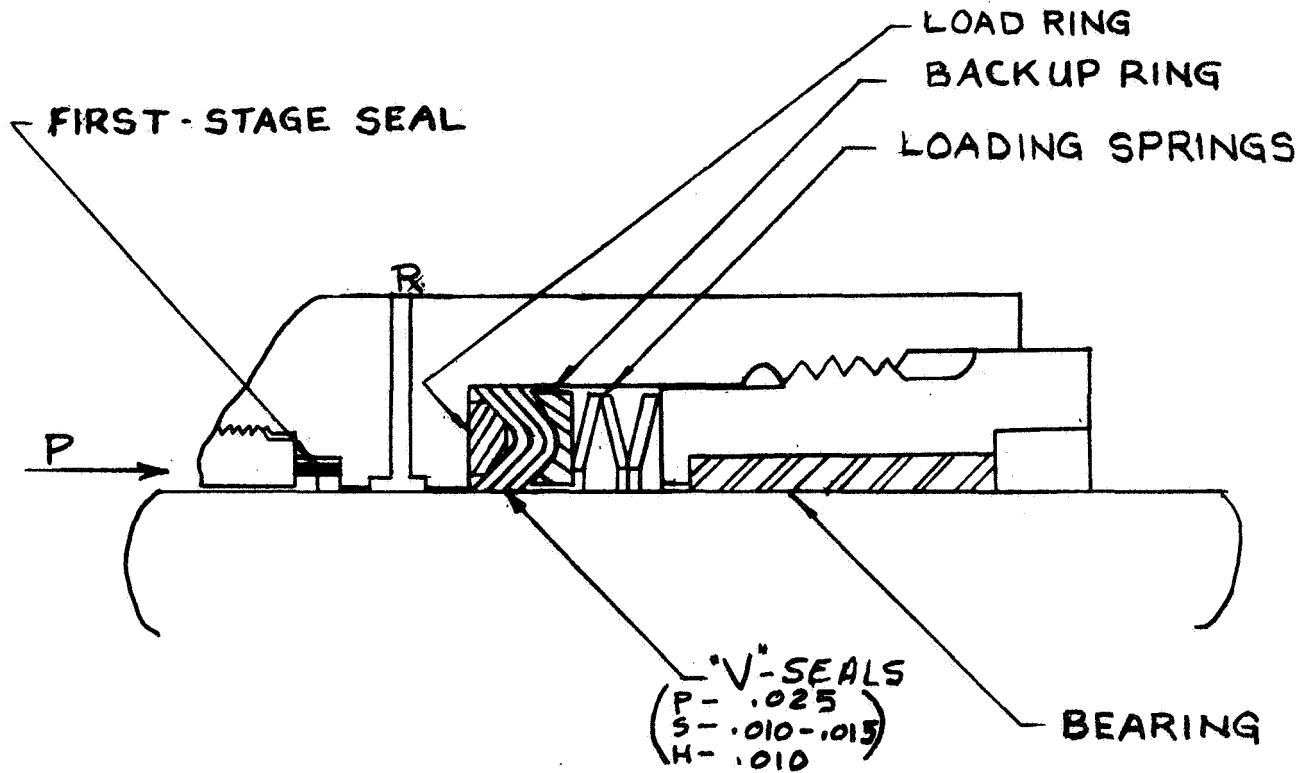


DESIGN AH - MULTIPLE REED SEAL-PLASTIC

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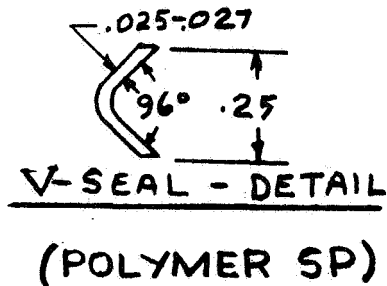


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DESIGN-B V-SEAL

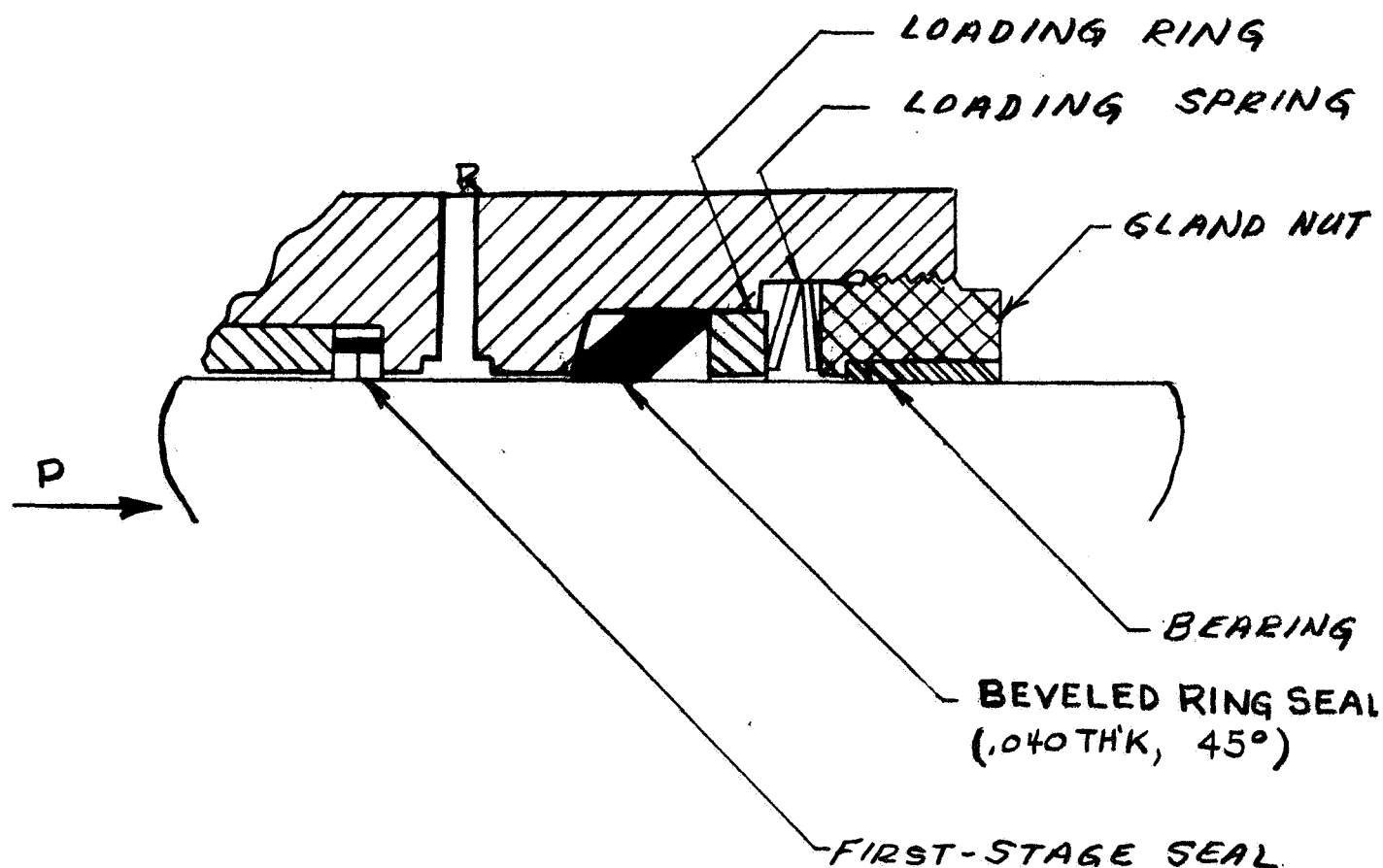
PLASTIC, SOFT METAL, HARD METAL



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DESIGN-C BEVELED RING SEAL - SOFT METAL

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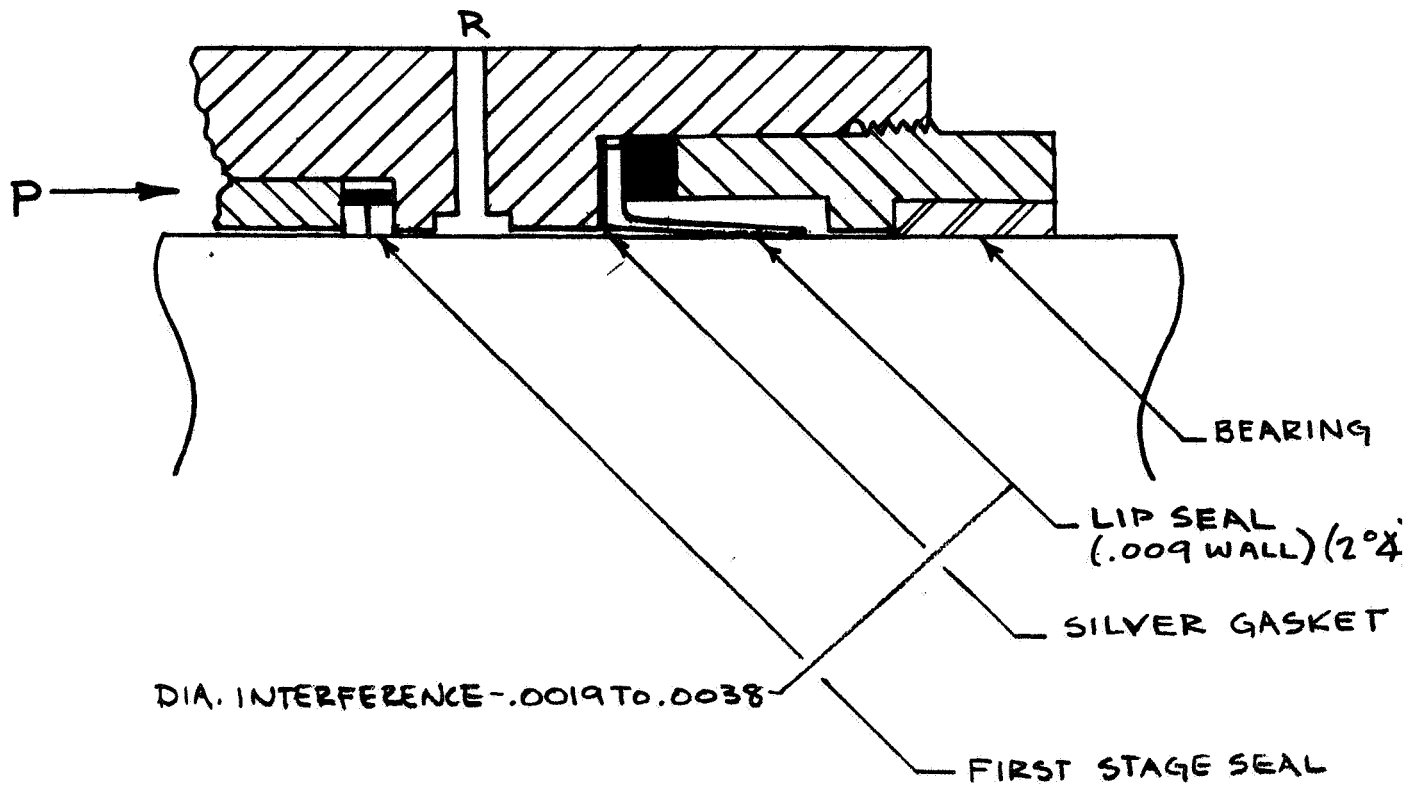
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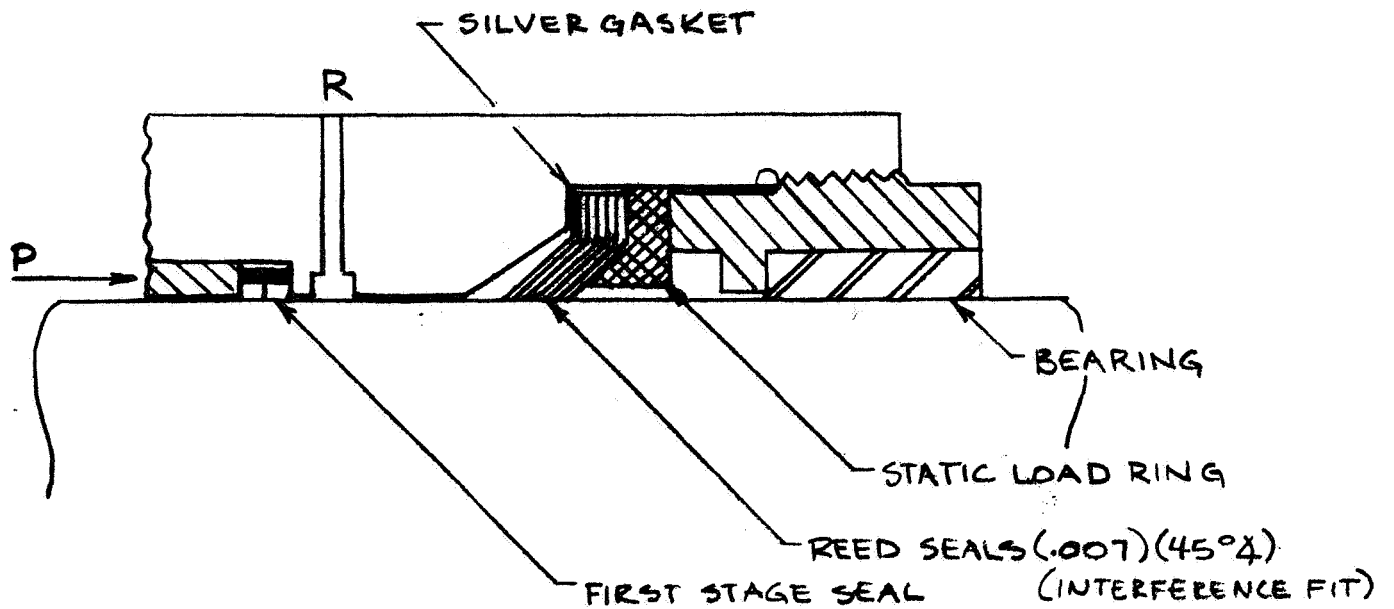


DESIGN D - HARD METAL LIP SEAL

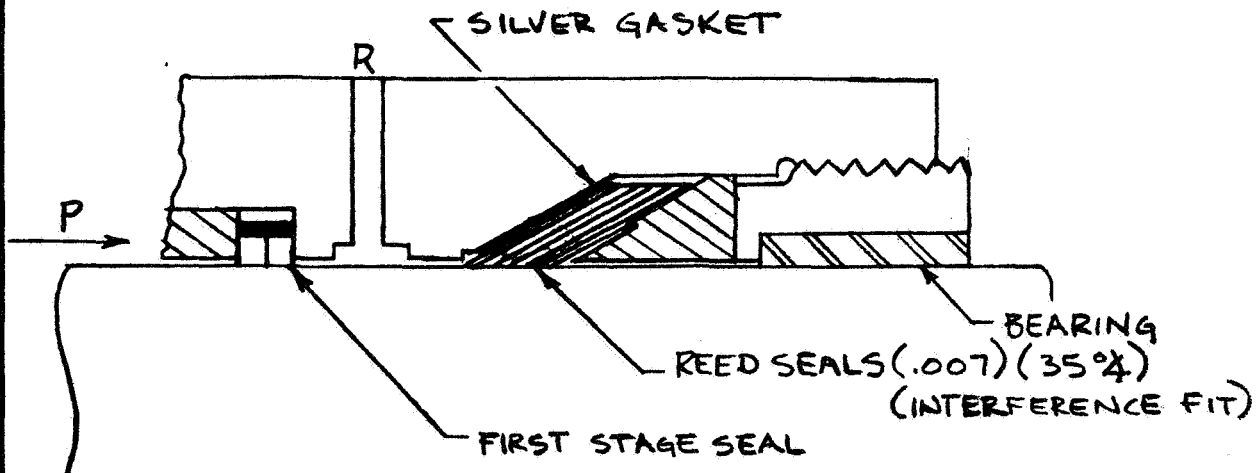
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DESIGN E - REED SEAL - HARD OR COMB. HARD AND SOFT METAL

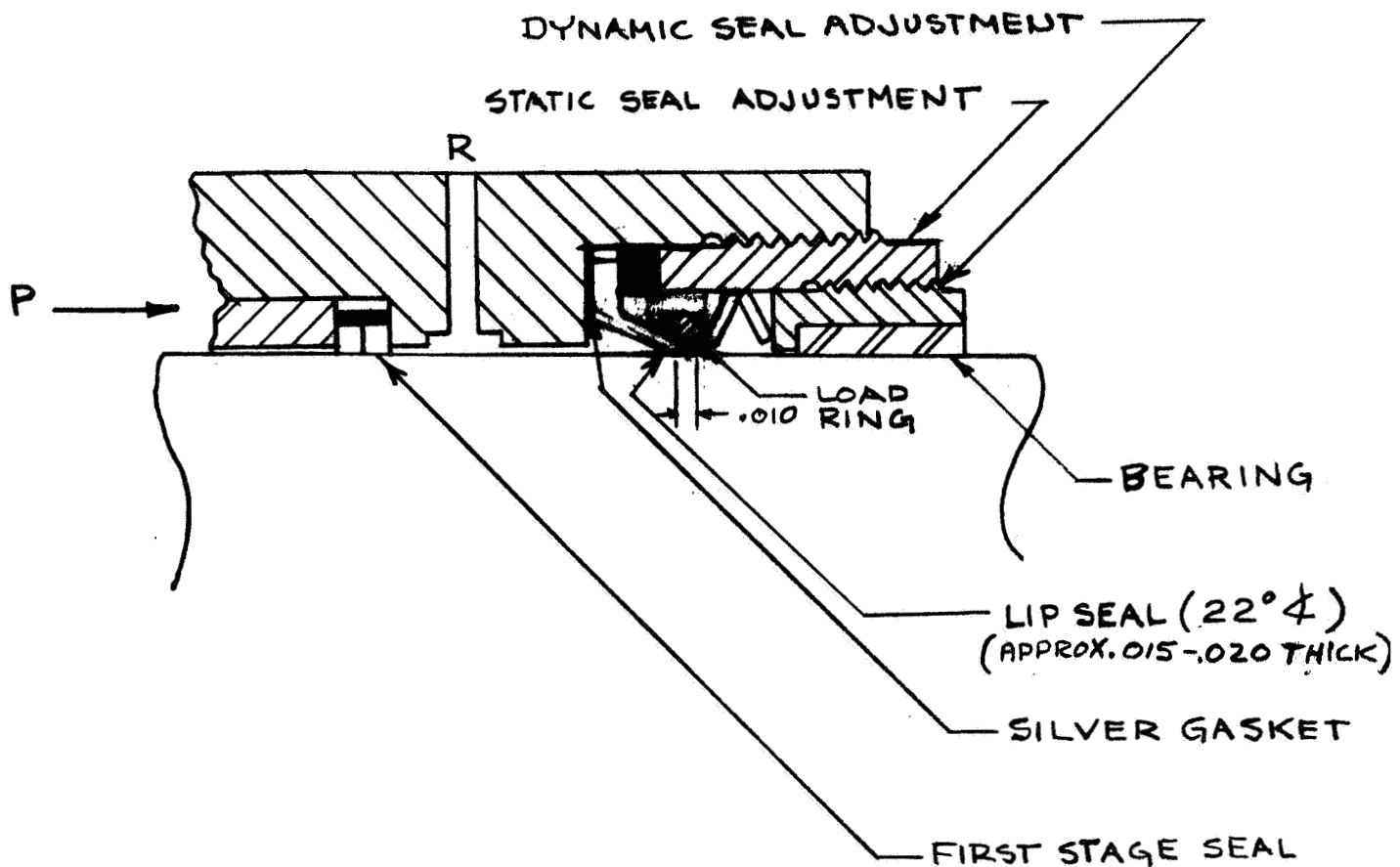


DESIGN H - REED SEAL - HARD OR COMB. HARD AND SOFT METAL

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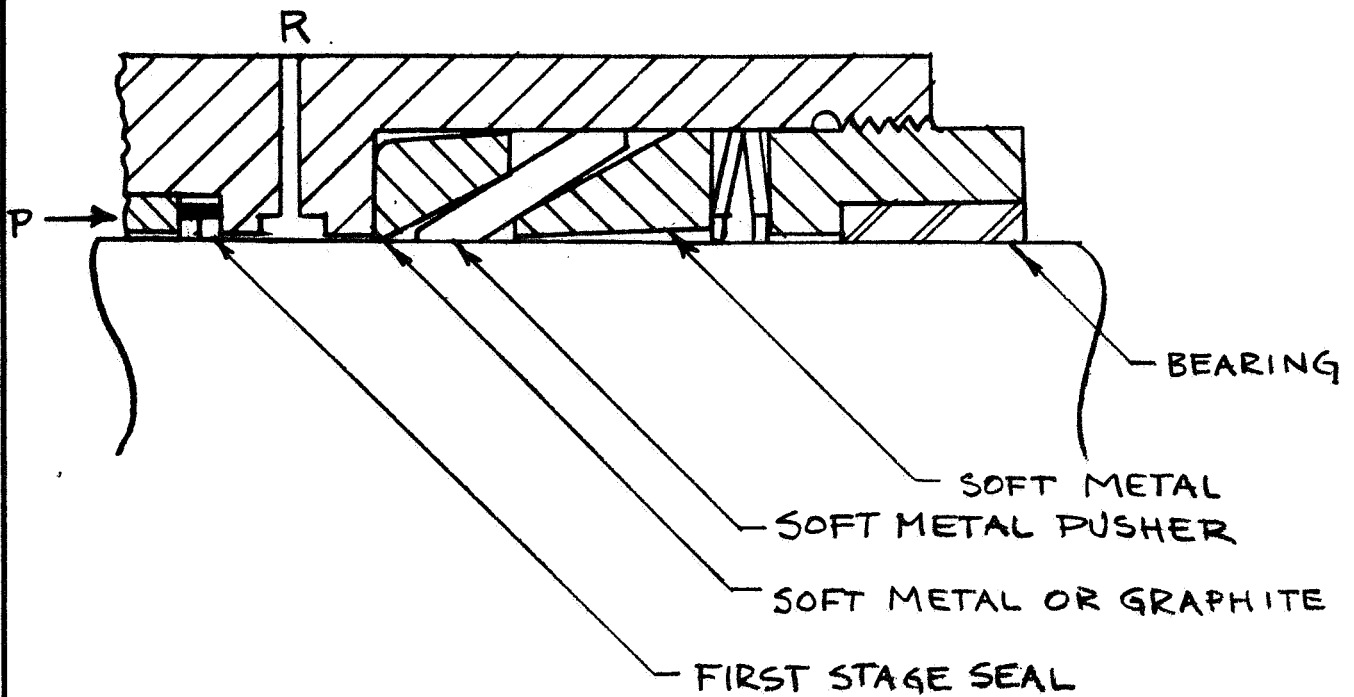


DESIGN F - HARD OR SOFT METAL LIP SEAL

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DESIGN G - DOUBLE WEDGE SEAL

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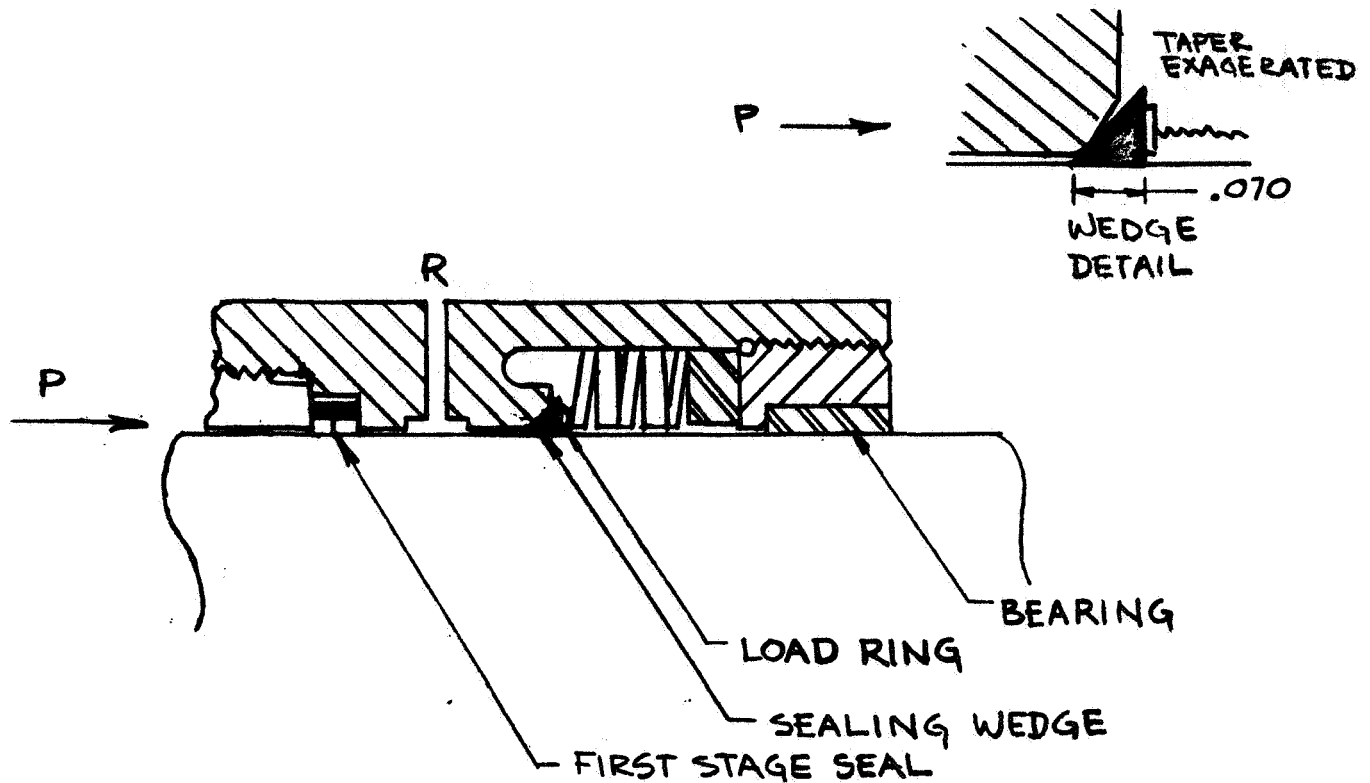
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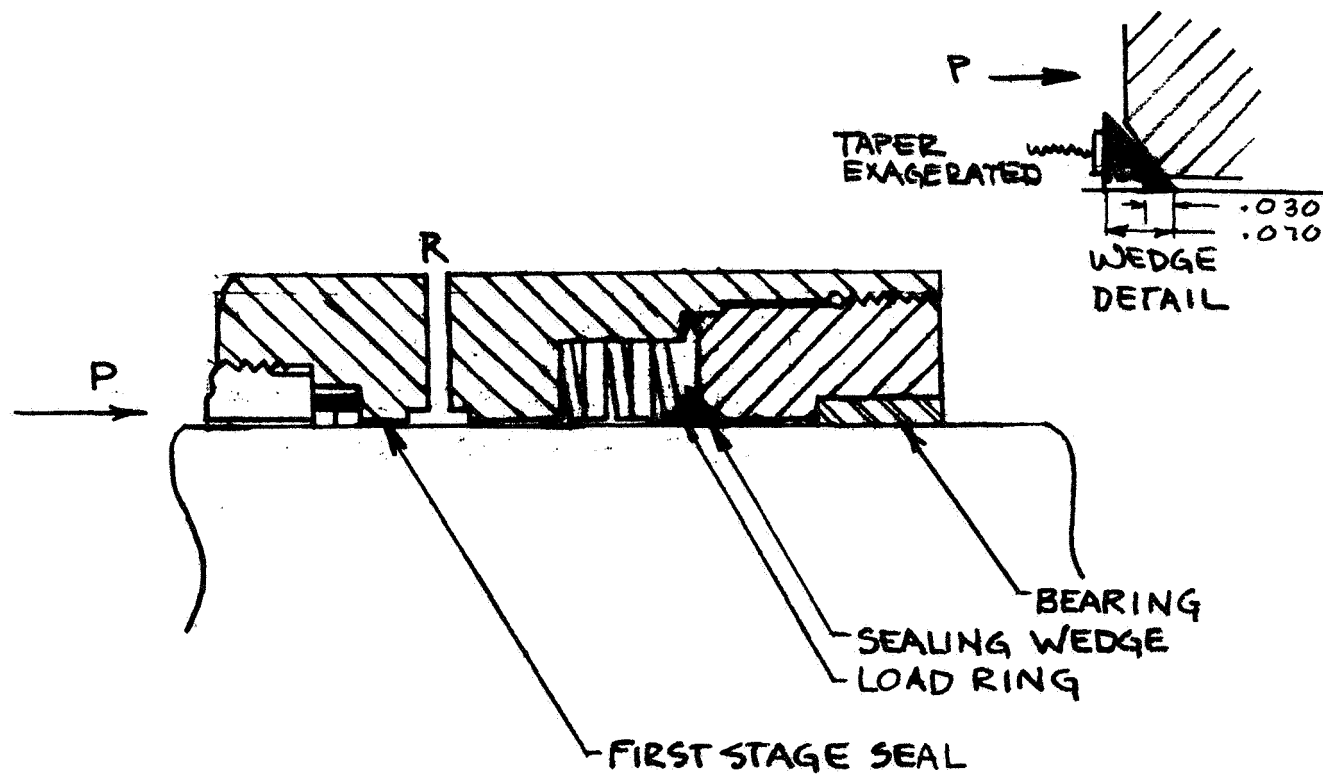


DESIGN I- WEDGE SEAL- SOFT METAL OR GRAPHITE

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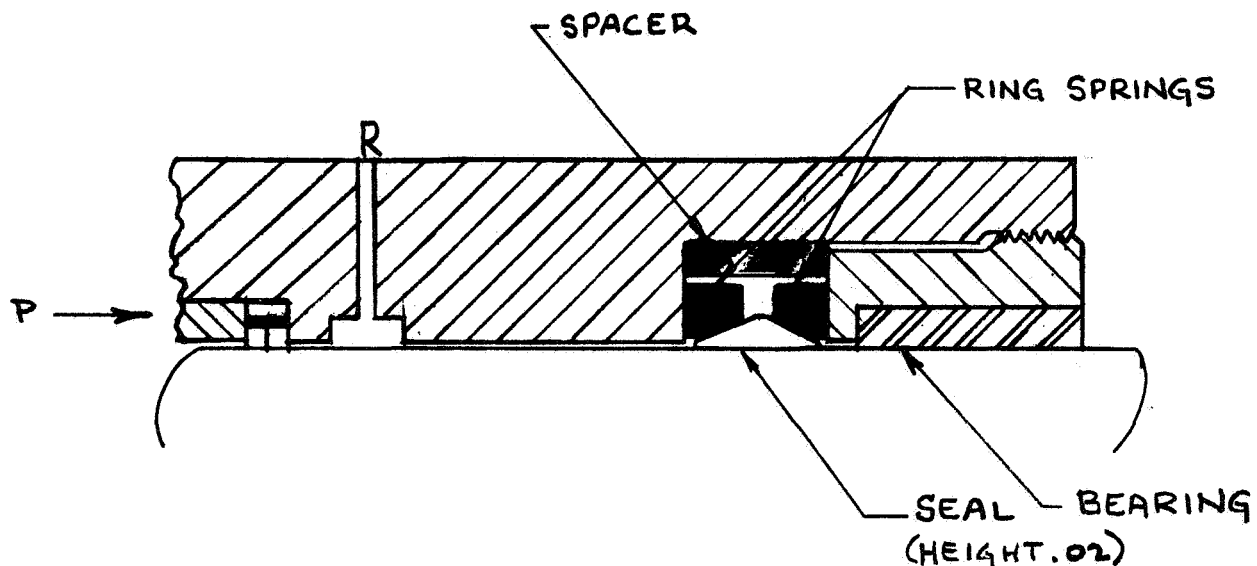


DESIGN IR - REVERSE WEDGE SEAL - SOFT METAL OR GRAPHITE

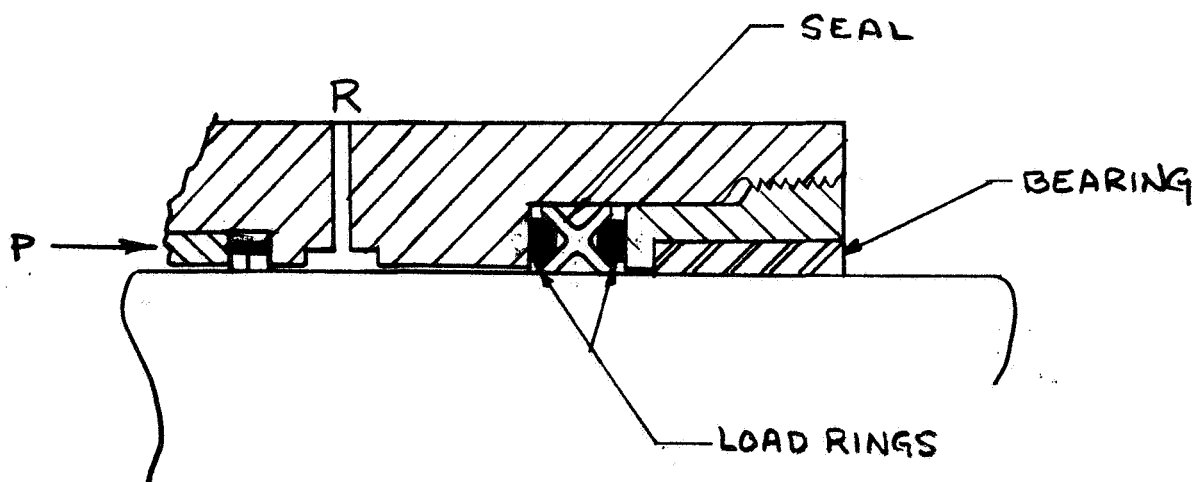
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DESIGN-J BOEING METALLIC RING SPRING SEAL -
HARDMETAL



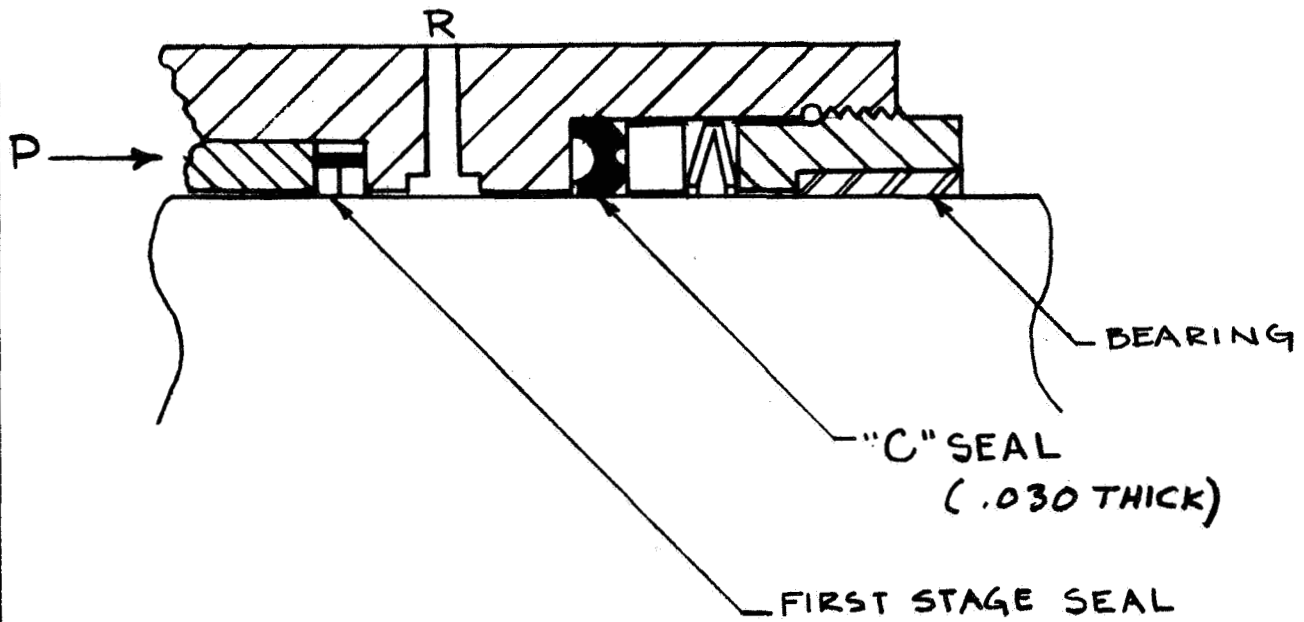
DESIGN-K METALLIC X-SEAL - HARD METAL

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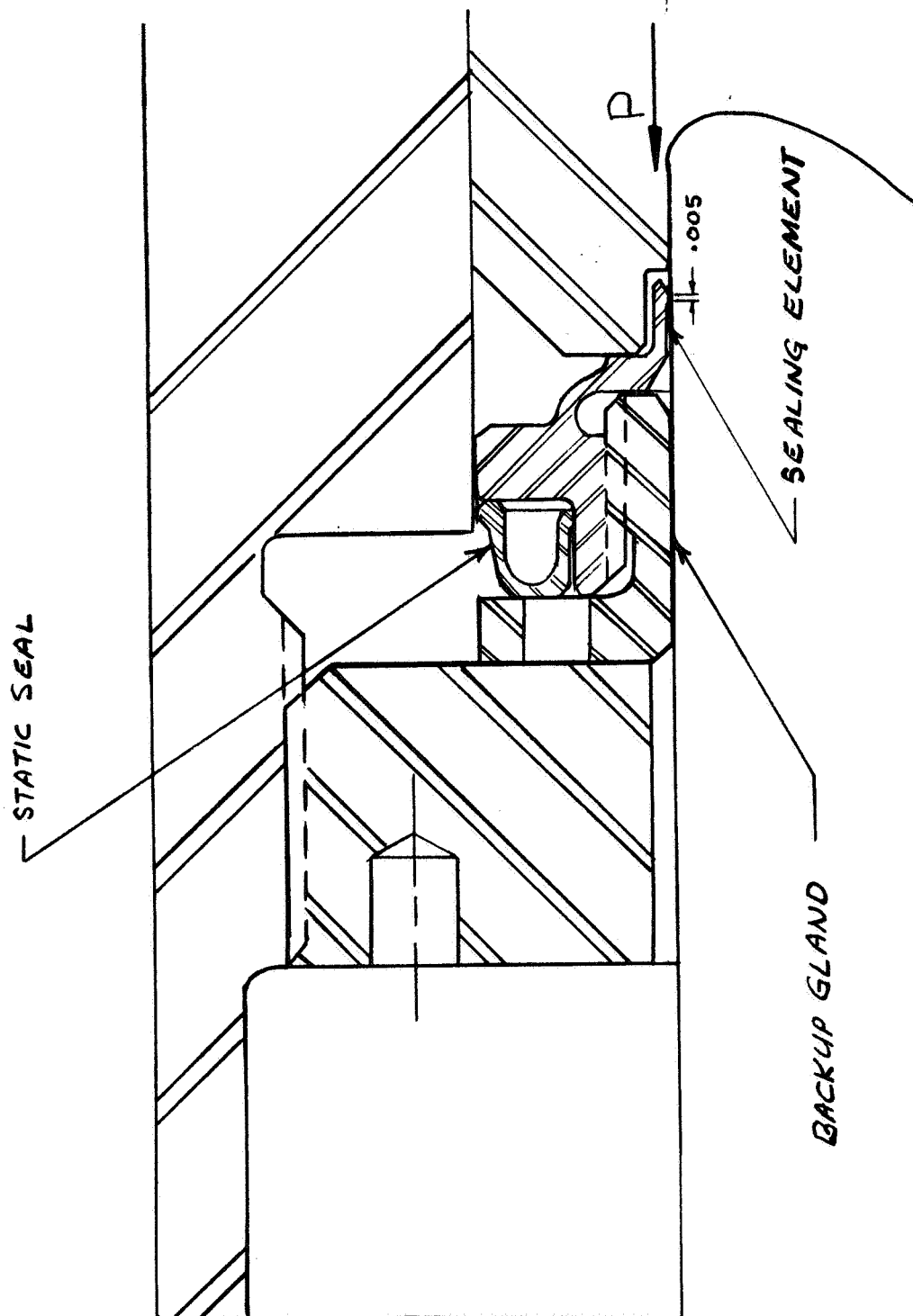


DESIGN L - METALLIC "C" SEAL

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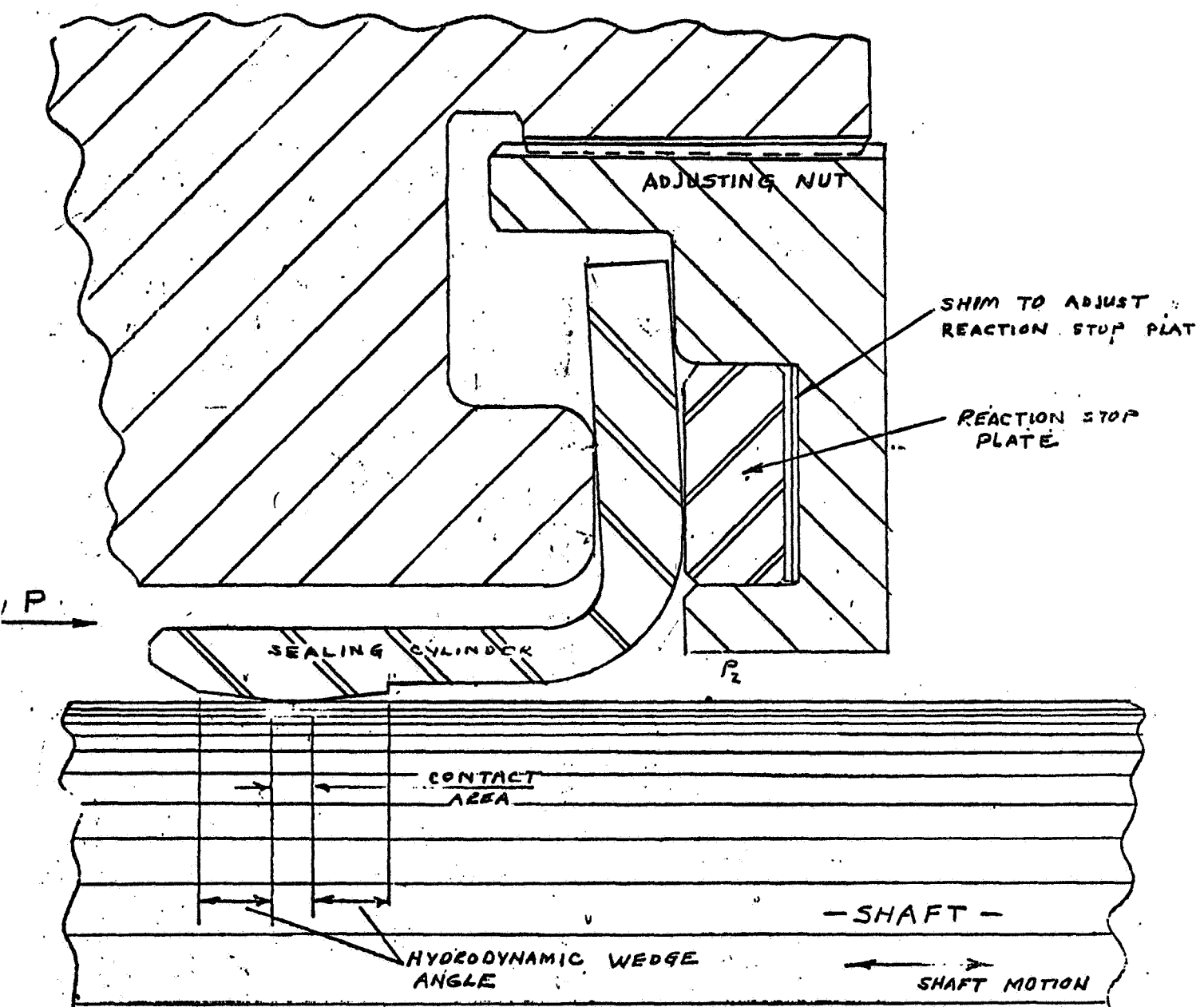
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by L.P. Ludwig DATE 8-30-65
CHKD. BY DATE

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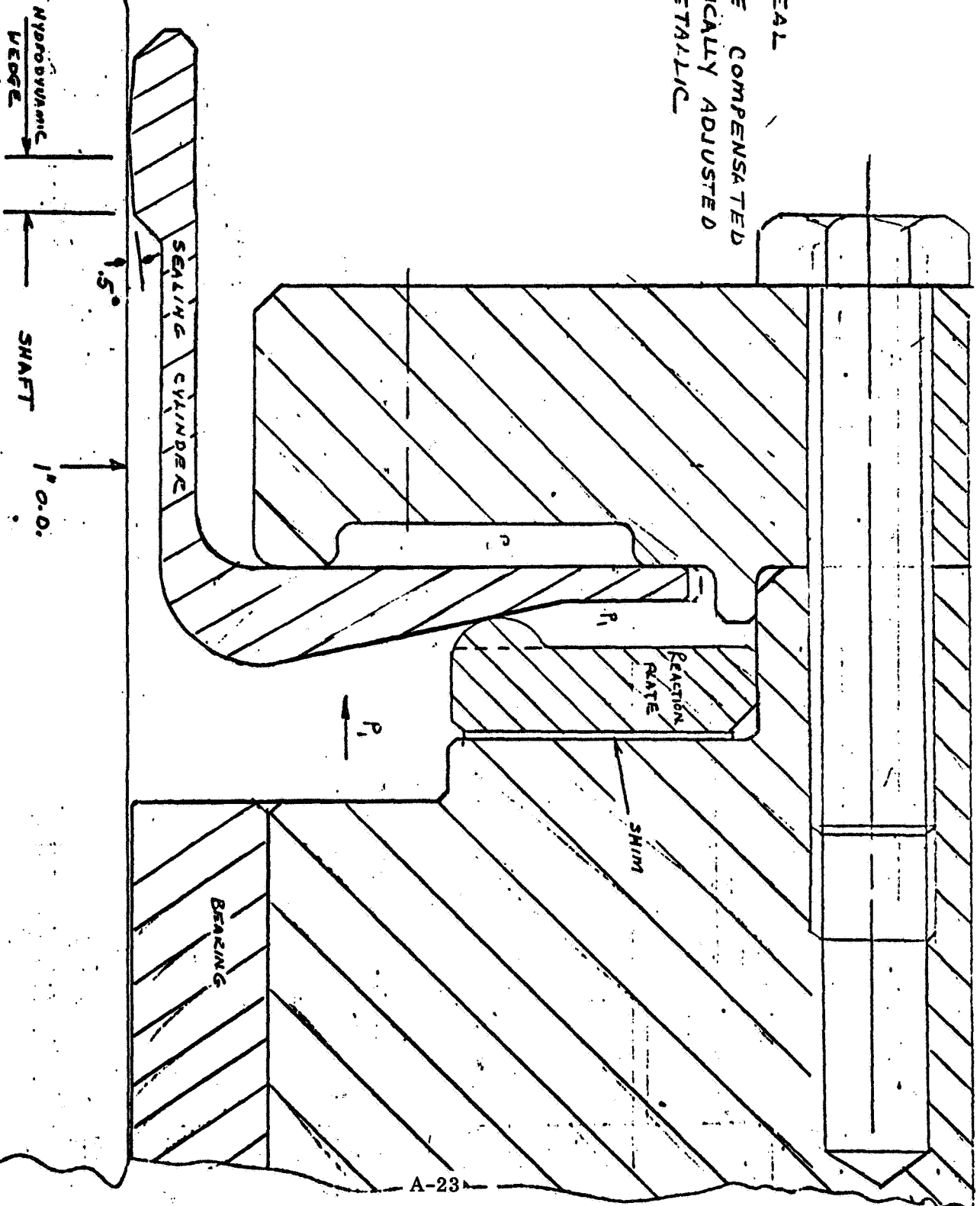
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William R. Morris 8-30-65

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DESIGN-O

ROD SEAL
PRESSURE COMPENSATED
MECHANICALLY ADJUSTED
ALL METALLIC



James P. Flaming 9-1-65

10 X SIZE

DESIGN A - REED SEAL - PLASTIC (POLYMER SP)

ESSENTIAL FACTORS		RATING	ADVANTAGES		DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded; predictable; controllable; good static sealing; pressure decreases leakage.	2		Pressure energizing reduces leakage. Static seal adjustment independent of dynamic seal. Behavior of Polymer SP predictable from experience in other configurations.		No dynamic load adjustment.
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to performance; pressure decreases wear.	2		Use of SP sheet material permits "stretch" compensation. Compensational ability recently demonstrated.		Pressure energizing may increase wear rate.
3. RELIABILITY Fails slowly and observably; rippled, resists abuse; redundant elements; resists pressure surges.	1		Capable of withstanding pressure surges. Material relatively resistant to foreign objects. Short term integrity of material well known.		No redundant elements. Long-term aging effects uncertain. High shear stresses may occur at edge of clamped portion.
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	2		Previous experience with material. Variables relatively easy to adjust and to evaluate		
DESIRABLE FACTORS					
1. DESIGN FEATURES Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.	1		Self-compensating. Hard chrome plate rod.		Coefficients of expansion mis-matched. Friction increases at higher pressure.
2. COST Easy to manufacture, machinable.	2		Simple - manufacture from sheet stock. No double concentricity problem - self centering.		Polymer SP somewhat difficult to hold to close tolerances.
3. SERVICEABILITY Easy to install, no spec. tool required. foolproof.	1		Omission of spacer will show up in pressure check. Interchangeable. Easy to install.		May require torque wrench to avoid overloading static lips. Requires care in "stretching" over rod and presetting of lips.
4. EXPERIENCE Past exp. with sim. approach.	1		Experience with SP V-seal (Design B) somewhat applicable. Limited experience with configuration A.		

COMMENTS: Rising friction vs. pressure characteristic - best suited for 2-stage flight control servos or utility servos. Consider multiple elements (see AH).
Sharp incidence angles unobtainable with sheet material - use of solid stock will sacrifice some flexibility.

SEAL PLATING - LOW PRESSURE SECOND-STAGE SEAL

DATE 12/65

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DESIGN AB - REED SEAL - PLASTIC (POLYMER SP)

ESSENTIAL FACTORS		RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded; predictable; controllable; good static sealing; pressure decreases leakage.	2	Pressure energizing reduces leakage. Static seal adjustment independent of dynamic seal. Behavior of polymer SP predictable from experience in other configurations.	No dynamic load adjustment. Several possible static leakage paths, but material tends to conform well.	
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to per- formance; pressure decreases wear.	2	Use of SP sheet material permits "stretch" compensation. Elements may transfer sealing function downstream as wear occurs, providing some increase in compensation life.	Pressure energizing may increase wear rate of upstream lips.	
3. RELIABILITY Fails slowly and observably; rugged, resists abuse; redundant elements; resists pressure surges.	2	Capable of withstanding pressure surges. Redundant elements. Material relatively resistant to foreign objects. Short term integrity of material is well known.	Long term aging effects uncertain.	
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	2	Previous experience with material. Variables relatively easy to adjust and to evaluate.		
DESIRABLE FACTORS				
1. DESIGN FEATURES Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.	1	Self compensating. Hard chrome plate rod.	Coefficients of expansion mismatched. Friction increases somewhat at higher pressure.	
2. COST Easy to manufacture, machinable.	2	Simple-manufacture from sheet stock. No double con- centricity problem-self centering.	Polymer SP somewhat difficult to hold to close tolerances.	
3. SERVICEABILITY Easy to install, no spec. tool required, foolproof.	1	Omission of spacer will show up in pressure checks. Interchangeable. Easy to install.	May require torque wrench to avoid overloading static lips. Some seal elements may be omitted without showing up in pressure checks. Requires presetting of lips.	
4. EXPERIENCE Fast exp. with sim. approach.	1	Experience with SP V-seal (Design E) somewhat applicable. Limited experience with configuration A.		

COMMENTS: Sharp incidence angles unobtainable with sheet stock - use of solid stock will sacrifice some flexibility. If friction vs. pressure characteristic can be controlled, this configuration is suited for single stage service.

SEAL RATING - LOW PRESSURE SECOND-STAGE SEAL
 DESIGN B - V-SEAL - PLASTIC (POLYMER SP)

DATE 12/65
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ESSENTIAL FACTORS		RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded; predictable; controllable; good static sealing; pressure decreases leakage.	2	Proven loading technique and static sealing. Behavior predictable from experience. Pressure energizing reduces high pressure leakage. Loading adjustable.	Static and dynamic loading are inter-related.	
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to per- formance; pressure decreases wear.	2	Demonstrated ability to compensate. Slight "stretch" fit can be employed. Elements transfer sealing function down- stream as wear occurs, providing some increase in com- pensation life.	Pressure energizing may increase wear rate of upstream lips.	
3. RELIABILITY Fails slowly and observably; rugged, resists abuse; redundant elements; resists pressure surges.	2	Redundant elements. Material relatively resistant to foreign objects. Short term behavior of material is fairly well known. Considerable cyclic life has been obtained with this configuration. Capable of withstanding surges.	Long term aging effects uncertain.	
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	2	Much data available - Configuration has been refined. Design and manufacturing techniques have been established.		
DESIRABLE FACTORS				
1. DESIGN FEATURES Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.	1	May be capable of some self-compensation if "stretch" is employed. Hard chrome plate rod.	Coefficients of expansion mismatched. Friction increases somewhat at higher pressures.	
2. COST Easy to manufacture, machinable.	1	Manufacturing techniques well established	Complex. Polymer SP somewhat difficult to hold to close tolerances. Must be made from solid stock. Close control of concentricities required.	
3. SERVICEABILITY Easy to install, no spec. tool required, foolproof.	1	Interchangeable. Reverse installation of seal elements not likely, and will show up in pressure check.	Omission of spacer or spring may not show up in pressure check. Can be installed in wrong sequence.	
4. EXPERIENCE Past exp. with sim. approach.	2	Extensive experience.		

COMMENTS: If friction vs. pressure characteristic can be controlled, this configuration is suited for single stage service.

SEAL RATING: - LOW PRESSURE SECOND-STAGE SEAL

DATE 12/65
REV

DESIGN B - V-SEAL - SOFT METAL

ESSENTIAL FACTORS		RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded; predictable; controllable; good static sealing; pressure decreases leakage.	2	Proven loading technique and static sealing. Behavior predictable from experience. Pressure energizing reduces high pressure leakage. Loading adjustable.	Static and dynamic loading are interrelated. Taper fairly critical in providing proper seat.	
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to per- formance; pressure decreases wear.	2	Compensation demonstrated. Limited downstream	Pressure energizing increases wear rate of upstream lips.	
3. RELIABILITY Fails slowly and observably; rugged, resists abuse; redundant elements; resists pressure surges.	2	Redundant elements. Short term behavior well known. Capable of withstanding pressure surges.		
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	2	Much data available - configuration has been refined. Design and manufacturing techniques have been established.		
DESIRABLE FACTORS				
1. DESIGN FEATURES Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.	1	Fairly well-matched coefficients of expansion. Hard chrome plate rod.	Friction increases at higher pressure. Limited "stretch" of soft metals precludes any degree of self compensation.	
2. COST Easy to manufacture, machinable.	1	Manufacturing techniques well established.	Complex. Close control of concentricities required.	
3. SERVICEABILITY Easy to install, no spec. tool required, foolproof.	1	Easy to install. Reverse installation of seal elements not likely, and will show up in pressure check.	Omission of spacer or spring may not show up in pressure check. Requires run-in.	
4. EXPERIENCE Past exp. with sim. approach.	2	Extensive experience.		

COMMENTS: Rising friction vs. pressure characteristic - best suited for 2-stage seal applications or utility services.

SEAL RATING - LOW PRESSURE SECOND-STAGE SEAL

DESIGN C - BEVELLED RING SEAL - SOFT METAL

DATE 12/66

REV

ESSENTIAL FACTORS		RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded; predictable; controllable; good static sealing; pressure decreases leakage.	1	See comment at bottom.		Behavior uncertain and difficult to predict. Effectiveness of static seal and load adjustment questionable.
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to per- formance; pressure decreases wear.	1	If inner edge seals statically, very small area is exposed to pressure.		Limited compensation because of geometric stability of design.
3. RELIABILITY Fails slowly and observably; rugged, resists abuse; redundant elements; resists pressure surges.	1	Multiple seal elements.		Principal is obscure and redundancy uncertain.
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	2	Relatively simple to evaluate		No data available.
DESIRABLE FACTORS				
1. DESIGN FEATURES Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.	1	Fairly well-matched coefficients of expansion. Low friction - small pressure - energizing effect. Probably can use hard chrome plate.		Limited "stretch" in soft metals precludes any degree of self-compensation.
2. COST Easy to manufacture, machinable.	1	Relatively easy to manufacture		Close concentricities required. Requires lapping in.
3. SERVICEABILITY Easy to install, no spec. tool required, foolproof.	1	Easy to install. No special tools required.		Backwards installation may not show up in pressure check. Requires run-in.
4. EXPERIENCE Past exp. with sim. approach.	0	None		

COMMENTS: A similar approach was once patented in England for application to high pressure air compressors, with good results claimed.

SEAL RATING - LOW PRESSURE SECOND-STAGE SEAL

DATE 12/85
REV

DESIGN F - LIP SEAL - SOFT METAL

ESSENTIAL FACTORS		RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded; predictable; controllable; good static sealing; pressure decreases leakage.	2		Static seal independent of dynamic seal. Dynamic load adjustable. Loading demonstrated in limited testing.	Stress levels at sealing lip somewhat difficult to predict.
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to performance; pressure decreases wear.	1		Wear appears to be low in limited testing (35000 cycles). Some compensation available.	Relationship of variables difficult to assess.
3. RELIABILITY Fails slowly and observably; rugged; resists abuse; redundant elements; resists pressure surges.	1		Load ring may protect against catastrophic failure under pressure surges.	No redundant elements. May not be capable of withstanding pressure surges without permanent set and increase in leakage.
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	2		Manufacturing techniques established. Limited testing has indicated feasibility of concept.	
DESIRABLE FACTORS				
1. DESIGN FEATURES Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.	1		Fairly well-matched coefficients of expansion.	Limited "stretch" in soft metals preclude any degree of self compensation.
2. COST Easy to manufacture, machinable.	1		All parts readily machinable.	Close tolerance requirement for proper load ring action.
3. SERVICEABILITY Easy to install, no spec. tool required. foolproof.	1		Unlikely to be installed backwards - will show up in pressure check.	Missing loading elements may not show up in pressure check. Requires run-in.
4. EXPERIENCE Past exp. with sim. approach.	1		Experience with V-seals somewhat applicable. Limited testing conducted on lip seal.	

COMMENTS

SEAL RATING - LOW PRESSURE SECOND-STAGE SEAL

DATE 12/85

REV

DESIGN G - DOUBLE WEDGE SEAL - SOFT METAL

ESSENTIAL FACTORS		RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded; predictable; controllable; good static sealing; pressure decreases leakage.	1	Adjustable.	Behavior difficult to predict.	
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to per- formance; pressure decreases wear.	1		Ability to compensate not apparent.	
3. RELIABILITY Fails slowly and observably; rugged, resists abuse; redundant elements; resists pressure surges.	1	Rugged seal element - not sensitive to damage by foreign object.	No redundant elements.	
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	1		Extensive development required. No data available.	
DESIRABLE FACTORS				
1. DESIGN FEATURES Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.				
2. COST Easy to manufacture, machinable.				
3. SERVICEABILITY Easy to install, no spec. tool required, foolproof.				
4. EXPERIENCE Past exp. with sim. approach.				

COMMENTS

SEAL RATING - LOW PRESSURE SECOND-STAGE SEAL

DESIGN 1 - WEDGE SEAL - SOFT METAL OR GRAPHITE

DATE 12/65
REV

ESSENTIAL FACTORS		RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded; predictable; controllable; good static sealing; pressure decreases leakage.	2	Demonstrated loading technique with good stress levels. Behavior can be predicted with reasonable accuracy. Load adjustable.		Static and dynamic loading interrelated.
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to performance; pressure decreases wear.	2	Demonstrated degree of compensation. Very small area exposed to pressure. May be capable of pressure balancing at some increase in exposed area.		
3. RELIABILITY Fails slowly and observably; rugged, resists abuse; redundant elements; resists pressure surges.	1	Seal element is fairly rugged and well supported. Not likely to fail catastrophically.		No redundant element. Preload necessary to withstand unseating by pressure surges plus rod friction may not be compatible with desired low-pressure friction characteristic.
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	1	Some test data available. Relatively easy to evaluate.		Experience shows that larger rod sizes (above 1") have excessive friction.
DESIRABLE FACTORS				
1. DESIGN FEATURES Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.	1	Fairly well-matched coefficients of expansion. Expansion level fairly constant due to small effect of pressure. Hard chrome plate rod.		Not self-compensating due to low "stretch" available.
2. COST Easy to manufacture; machinable.	1	Fairly easy to manufacture. Self-centering.		Diametral tolerances fairly critical. Excessive bearing clearances may cause poor seating.
3. SERVICEABILITY Easy to install, no spec. tool required, foolproof.	2	Easy to install. Improper installation will probably show up in pressure check.		Requires run-in, particularly with soft metal.
4. EXPERIENCE Fast exp. with sim. approach.	2	Experience documented.		

COMMENTS: Graphite exhibits greater compressibility (higher $\frac{S}{E}$) than silver.

SEAL RATING: - LOW PRESSURE SECOND-STAGE SEAL
 DESIGN IR - REVERSE WEDGE SEAL - SOFT METAL OR GRAPHITE

DATE 12/85
 REV

ESSENTIAL FACTORS		RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded; predictable; controllable; good static sealing; pressure decreases leakage.	1	1	Loading technique demonstrated to some extent in Design 1. Behavior reasonably predictable. Pressure has slight tendency to decrease leakage.	Requires shimming for load adjustment. Static and dynamic loading interrelated. Requires extra static seal. Taper fairly critical in providing proper seat.
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to per- formance; pressure decreases wear.	2	2	Design 1 has demonstrated a degree of compensation. Relatively small unbalanced area acted on by pressure. Capable of being balanced to considerable degree.	Pressure has slight tendency to increase wear.
3. RELIABILITY Fails slowly and observably; rugged, resists abuse; redundant elements; resists pressure surges.	2	2	Seal element fairly rugged and well supported. Not likely to fail catastrophically. Capable of withstanding pressure surges.	No redundant element.
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	1	1	Limited data available from Design 1. Relatively easy to evaluate.	May exhibit same size limitation as Design 1.
DESIRABLE FACTORS				
1. DESIGN FEATURES Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.	1	1	Fairly well-matched coefficients of expansion. Friction level fairly constant due to small unbalanced area. Hard chrome plate rod.	Not self-compensating due to low "stretch" available.
2. COST Easy to manufacture, machinable.	1	1	Fairly easy to manufacture. Self-centering	Diametral tolerances fairly critical. Excessive bearing clearances may cause poor seating, but bearing is piloted on same piece as seat.
3. SERVICEABILITY Easy to install, no spec. tool required, foolproof.	2	2	Easy to install. Improper installation will probably show up in pressure check.	Requires run-in, particularly with soft metal.
4. EXPERIENCE Past exp. with sim. approach.	2	2	Some experience with Design 1 applicable.	

COMMENTS: Graphite exhibits greater compressibility (higher S/E) than silver.

SEAL RATING - LOW PRESSURE SECOND-STAGE SEAL

DESIGN L - "C" SEAL - SOFT METAL

DATE 12/65
REV

ESSENTIAL FACTORS		RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded; predictable; controllable; good static sealing; pressure decreases leakage.	2		Reasonably predictable behavior. Pressure energizing tends to reduce leakage at high pressure. Load adjustable.	Geometry limits loading ability; static and dynamic loading interrelated.
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to per- formance; pressure decreases wear.	1		Some degree of compensation demonstrated in high pressure application.	Pressure energizing tends to increase wear at high pressure. Stable geometry may limit compensation.
3. RELIABILITY Fails slowly and observably; resists abuse; redundant elements; resists pressure surges.	1		Capable of withstanding pressure surges.	No redundant element.
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	2		Some test data available. Relatively easy to evaluate.	
DESIRABLE FACTORS				
1. DESIGN FEATURES Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.	1		Fairly well-matched coefficients of expansion.	Friction increases at higher pressure. Not self-compensating because of limited "stretch" of soft metal.
2. COST Easy to manufacture, machinable.	1		Fairly simple configuration.	Double concentricities must be held closely.
3. SERVICEABILITY Easy to install, no spec. tool required, foolproof.	1		Easy to install. Omission of spring or spacer will probably show up in pressure check.	Reverse installation may not show up in pressure check. Requires run-in.
4. EXPERIENCE Past exp. with sim. approach.	1		Some experience.	

COMMENTS: Optimum low pressure characteristic may not be suitable for high pressure application.

SEAL RATING - LOW PRESSURE SECOND-STAGE SEAL
 DESIGN B - V-SEAL - HARD METAL

DATE 12/85
 REV

ESSENTIAL FACTORS		RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded; predictable; controllable; good static sealing; pressure decreases leakage.	2		Loading technique and static sealing proven with other materials. Behavior reasonably predictable. Pressure energizing reduces high pressure leakage. Load adjustable.	Static and dynamic loading are interrelated.
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to performance; pressure decreases wear.	1		Compensation with "stretch" fit. Downstream transfer of sealing function may occur as wear progresses.	Wear rate may be high for thin lip and high bearing load. Pressure energizing may increase wear rate.
3. RELIABILITY Fails slowly and observably; rugged; resists abuse; redundant elements; resists pressure surges.	1		Redundant elements. Capable of withstanding pressure surges.	Sealing lips susceptible to damage by foreign objects.
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	2		Previous experience with configuration generally applicable.	
DESIRABLE FACTORS				
1. DESIGN FEATURES Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.	1		Well matched coefficient of expansion. Self-compensating due to "stretch".	May require flame plated rod. Friction may be high because of high bearing loads and pressure energizing.
2. COST Easy to manufacture, machinable.	0			V-shape fairly expensive to machine in thin sections. Complex. Close control of concentricities required.
3. SERVICEABILITY Easy to install, no spec. tool required, foolproof.	0		Reverse installation of seal elements not likely and will show up in pressure check.	Tends to jam in cavity due to high bearing loads and permanent set. Requires care in "stretching" over rod. Omission of element may not show up in pressure check. Requires run-in.
4. EXPERIENCE Past exp. with sim. approach.	2		Extensive experience with V-seals.	

COMMENTS: Rising friction vs. pressure characteristic.

SEAL RATING - LOW PRESSURE SECOND-STAGE SEAL

DESIGN D - LIP SEAL - HARD METAL

DATE 12/85
REV

ESSENTIAL FACTORS		RATING	ADVANTAGES		DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded; predictable; controllable; good static sealing; pressure decreases leakage.	2		Demonstrated loading technique and static sealing. Reasonably predictable behavior. Static seal independent of dynamic seal.		Slight leakage inherent in pressure relief behavior. Load not adjustable.
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to per- formance; pressure decreases wear.	2		Low wear rate - no thin wear section and pressure relief Wear probably gradual. Compensation inherent due to "stretch".		Increased leakage with wear due to lower contact pressure (increased contact area).
3. RELIABILITY Fails slowly and observably; rugged, resists abuse; redundant elements; resists pressure surges.	1		See above. Not overly sensitive to damage by foreign objects.		No redundant elements. May not be capable of withstanding pressure surges without backup.
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	2		Configuration previously tested for short time. Relatively easy to evaluate.		
DESIRABLE FACTORS					
1. DESIGN FEATURES -Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.	2		Well-matched coefficient of expansion. Self-compensating due to "stretch". Hard chrome plate rod. Low friction due to pressure relief.		May require flame plated rod for best wear life.
2. COST Easy to manufacture, machinable.	1		Simple, fairly easy to manufacture. No double concentricity problem - self-centering. Rough drawing may be feasible in production.		Close tolerance on wall. Taper may be critical.
3. SERVICEABILITY Easy to install, no spec. tool required, foolproof.	2		Difficult to install improperly. Taper fit eases problem of "stretching" over rod. Improper installation will probably show up in pressure check.		Requires run-in.
4. EXPERIENCE Past exp. with sim. approach.	2		See (4) above.		

COMMENTS: May be suitable for first stage seal, or single stage seal if friction is not critical.

SEAL RATING - LOW PRESSURE SECOND-STAGE SEAL

DESIGN E & H - REED SEAL - HARD METAL

DATE 12/65

REV

ESSENTIAL FACTORS		RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded; predictable; controllable; good static sealing; pressure decreases leakage.	2	Pressure energizing reduces high pressure leakage. Approach has been demonstrated in other programs. Static and dynamic adjustments independent.	Several possible static leakage paths, but high clamping forces are available. No dynamic load adjustment.	
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to performance; pressure decreases wear.	1	Compensation fair with "stretch" fit. Elements transfer sealing function downstream as wear occurs, providing some increase in compensation life.	Wear rate of individual lips may be high for thin lip and high bearing loads. Pressure energizing may increase wear rate of upstream lips.	
3. RELIABILITY Fails slowly and observably; rugged, resists abuse; redundant elements; resists pressure surges.	2	Capable of withstanding pressure surges. Redundant elements.	Sealing lips may be susceptible to damage by foreign objects.	
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	2	Previous experience with configuration documented.		
DESIRABLE FACTORS				
1. DESIGN FEATURES Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.	1	Well-matched coefficient of expansion. Self-compensating due to "stretch".	May require flame plated rod. Friction may be high because of high bearing loads and pressure energizing.	
2. COST Easy to manufacture, machinable.	2	Fairly simple and easy to manufacture. No double concentricity problem - self centering. Rough drawing may be feasible in production.		
3. SERVICEABILITY Easy to install, no spec. tool required, foolproof.	1	Fairly easy to install thin lips relatively easy to "stretch" over rod.	Some seal elements may be omitted without showing up in pressure check. Requires run-in. May require torque wrench to avoid overloading static lips.	
4. EXPERIENCE Last exp. with sim. approach.	2	Previous experience with configuration.		

COMMENTS: Rising friction vs. pressure characteristic - best suited for 2-stage flight control services or utility services.

SEAL RATING - LOW PRESSURE SECOND-STAGE SEAL

DESIGN F - LIP SEAL - HARD METAL

DATE 12/85
REV

ESSENTIAL FACTORS		RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM	Conformable; readily loaded; predictable; controllable; good static sealing; pressure decreases leakage.	2	Static sealing inherent. Individual adjustment of static and dynamic seals. Load adjustable.	Stress levels at sealing lip somewhat difficult to predict.
2. WEAR & COMPENSATION	Low wear or able to compensate; wear not detrimental to performance; pressure decreases wear.	1	Compensation may be inherent due to "stretch".	Relationship of bearing loads to wear rate, friction and leakage uncertain.
3. RELIABILITY	Fails slowly and observably; rugged; resists abuse; redundant elements; resists pressure surges.	1	Load ring may protect against catastrophic failure under pressure surges.	No redundant elements. Lip somewhat sensitive to damage by foreign objects.
4. SHORT TERM POTENTIAL	Knowledge available or can be acquired in reasonable time.	1	Previous experience with design "D".	Fairly difficult to control parameters, particularly thin lip of seal.
DESIRABLE FACTORS				
1. DESIGN FEATURES	Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.	1	Well-matched coefficient of expansion. Self-compensating due to "stretch". Some pressure relief may be available to reduce friction.	May require flame plated rod.
2. COST	Easy to manufacture, machinable.	1	All parts readily machineable.	Close tolerance-requirements on thin lips.
3. SERVICEABILITY	Easy to install, no spec. tool required. foolproof.	1	Unlikely to be installed backwards - likely to show up in pressure check.	Missing loading elements may not show up in pressure check. Requires run-in.
4. EXPERIENCE	Past exp. with sim. approach.	1	Previous experience with somewhat similar configuration.	

COMMENTS

SEAL RATING - LOW PRESSURE SECOND-STAGE SEAL
 DESIGN J - BOEING RING SPRING SEAL - HARD METAL

DATE 12/65
 REV

ESSENTIAL FACTORS		RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded; predictable; controllable; good static sealing; pressure decreases leakage.	1		Demonstrated initial loading ability.	Load difficult to maintain. No positive static seal - pressure may relax spring load, permitting increased leakage.
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to per- formance; pressure decreases wear.	1		Wear likely to be uniform.	High forces required for compensation, and limited by geometry or ring spring.
3. RELIABILITY Fails slowly and observably; rugged, resists abuse; redundant elements; resists pressure surges.	2		Resistant to pressure surges. Rugged element. Seal element resists damage by foreign objects.	No redundant elements.
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	1			Extensive development required. Proprietary to Boeing - may be impractical to develop within program scope.
DESIRABLE FACTORS				
1. DESIGN FEATURES Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.	1		Well matched coefficient of expansion	Requires flame plated rod. Friction may be high because of high bearing load.
2. COST Easy to manufacture, machinable.	1		Cannot be installed backwards. Missing elements will show	Fairly expensive because of close tolerance requirements on all three members.
3. SERVICEABILITY Easy to install, no spec. tool required. foolproof.	1		Cannot be installed backwards. Missing elements will show up in pressure check.	Requires run-in.
4. EXPERIENCE Past exp. with sim. approach.	1		Previous experience with seal in 1/2-in. rod size.	

COMMENTS

SUMMARY - LOW PRESSURE SECOND-STAGE SEAL

DATE 12/85
REV

DESIGN K - X-Seal - Hard Metal

ESSENTIAL FACTORS		RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM	Conformable; readily loaded; predictable; controllable; good static sealing; pressure decreases leakage.	1	Static sealing good. Some load adjustment inherent.	Experience has shown this design to be very difficult to control as a dynamic seal. Static and dynamic loading interrelated.
2. WEAR & COMPENSATION	Low wear or able to compensate; wear not detrimental to performance; pressure decreases wear.	1	"Stretch" fit may augment loading for compensation.	See above - high wear rate can occur if precise control of variables is not attained.
3. RELIABILITY	Fails slowly and observably; rugged; resists abuse; redundant elements; resists pressure surges.	1	Resists pressure surges.	See above. Foreign objects may damage rod or lips. Difficult to obtain uniform loading necessary for redundancy.
4. SHORT TERM POTENTIAL	Knowledge available or can be acquired in reasonable time.	1		Proprietary item requiring extensive development in conjunction with vendor.
DESIRABLE FACTORS				
1. DESIGN FEATURES	Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.			
2. COST	Easy to manufacture, machinable.			
3. SERVICEABILITY	Easy to install, no spec. tool required, foolproof.			
4. EXPERIENCE	Fast exp. with sim. approach.			

COMMENTS

SEAL RATING - LOW PRESSURE SECOND-STAGE SEAL
 DESIGN M - BFG SEAL - HARD METAL

DATE 12/65
 REV

ESSENTIAL FACTORS		RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded; predictable; controllable; good static sealing; pressure decreases leakage.	2		Demonstrated loading ability. Behavior predictable to some extent through experience. Pressure energizing decreases leakage.	Static & dynamic seal loaded together. Loading difficult to control and adjust.
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to performance; pressure decreases wear.	1		Compensation inherent due to "stretch".	Wear rate may be high for small sealing lip and hard material. Pressure surges can cause high contact stresses at sealing surface.
3. RELIABILITY Fails slowly and observably; rugged, resists abuse; redundant elements; resists pressure surges.	1		Not likely to fail catastrophically because of backup gland.	No redundant element. Foreign objects may damage lip or rod.
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	1		Some experience.	Complex proprietary item requiring development in conjunction with vendor.
DESIRABLE FACTORS				
1. DESIGN FEATURES Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.	1		Coefficients of expansion well matched. Some self-compensation if "stretched".	May require flame plated rod, according to vendor. Friction increases at higher pressure.
2. COST Easy to manufacture, machinable.	0			Complex and expensive. Multiple concentricities and concentric surfaces for static seal are critical.
3. SERVICEABILITY Easy to install, no spec. tool required, foolproof.	1		Not likely to be installed backwards, and would show up in pressure check.	Static seal can be installed backwards, but may show up in pressure check. Requires run-in.
4. EXPERIENCE Past exp. with sim. approach.	2		Tested at lower (300°F) temp. by vendor.	

COMMENTS

SEAL RATING - LOW PRESSURE SECOND-STAGE SEAL

DATE 12/85

DESIGN N - (NASA DESIGN 2) - HARD METAL

REV

O - (NASA DESIGN 5)

ESSENTIAL FACTORS		RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM	Conformable; readily loaded; predictable; controllable; good static sealing; pressure decreases leakage.	1	Loading similar to Design M. Pressure has little effect on leakage for wall thickness shown.	Static and dynamic seal loading interrelated. Loading difficult to control and adjust - requires shimmming to final adjustment.
2. WEAR & COMPENSATION	Low wear or able to compensate; wear not detrimental to performance; pressure decreases wear.	2	Compensation inherent in loading. Pressure has little effect on wear.	Wear rate may be high initially for line contact area and hard material. Hydrodynamic effect on outstrokes questionable for low rod speeds. No effective hydrodynamic action on instrokes.
3. RELIABILITY	Fails slowly and observably; rugged; resists abuse; redundant elements; resists pressure surges.	1	Not likely to fail catastrophically because of reaction stop-plate.	No redundant element. Foreign objects may damage rod.
4. SHORT TERM POTENTIAL	Knowledge available or can be acquired in reasonable time.	1	Similar to Design M.	Complex and sensitive geometry for loading.
DESIRABLE FACTORS				
1. DESIGN FEATURES	Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.	2	Coefficients of expansion well matched. Self compensating. Pressure has little effect on friction.	May require flame plated rod (similar to Design M).
2. COST	Easy to manufacture, machinable.	1	Fairly simple construction.	Squareness of mating parts critical.
3. SERVICEABILITY	Easy to install, no spec. tool required, foolproof.	1	Not likely to be installed backwards, and would show up in pressure check.	Requires trial and error shimmming. Probably requires run-in and readjustment.
4. EXPERIENCE	Past exp. with sim. approach.	1	Some experience with similar design.	

COMMENTS May be suited to non-control system high pressure applications.

APPENDIX B

EXHIBIT "A"

High Temperature Hydraulic System Actuator Seals

The Contractor shall furnish the necessary personnel, facilities, services and materials and otherwise do all things necessary for, or incident to, the work described below:

The work to be performed shall provide for the investigation of materials and designs of seals for potential use with hydraulic fluids in advanced supersonic aircraft. This investigation shall be directed to seals intended to function efficiently and reliably for 3000 hours in the temperature range -40°F to 600°F.

TASK I - Apparatus for Evaluation of Seal Materials and Designs

- A. Facilities shall be provided for the measurement of hardness, elasticity, mechanical strength and other mechanical properties important to hydraulic system seals.
- B. Facilities shall be provided in which seal materials can be evaluated at temperatures of 400°, 500°, and 600°F for chemical compatibility with five fluids to be selected by the Contractor with the approval of the NASA Project Manager. Periods of 150 hours shall be used. The fluids shall be degassed and compatibility shall be established in an inerted atmosphere system. Inerting shall be accomplished with 99.99 percent by volume nitrogen having an oxygen content of not more than 50 ppm, a hydrocarbon content (as methane) of not more than 50 ppm, and a dew point of -90°F or lower.
- C. Facilities shall be provided for a rod seal test unit using one-inch diameter seals at 100 psi. Operation shall alternate between operation at 30-40 CPM with ± 2 to ± 4 inch stroke and 100-300 CPM with $\pm .05$ to $\pm .10$ inch stroke to simulate maneuvering and auto-pilot inputs to the actuator. The fluid temperature levels shall be 400°F, 500°F, and 600°F with the temperature for the seal in the actuator unit no less than the fluid temperature. Leakage and actuator forces shall be measured.
- D. Facilities shall be provided for a rod seal test unit using seals of 1 inch and 3 inch diameters at pressures from 0 to 4000 psi. Operation shall include a cycling rate of 15-20 CPM with stroke length alternating from strokes ($\pm \frac{1}{2}$ to ± 1 inch) to strokes (± 2 to ± 4 inch). Operation shall also include a cycling rate of 100-300 CPM with a constant stroke length ($\pm .05$ to $\pm .10$ inch). The fluid temperature level shall be 500°F and the temperature of the seal in the actuator unit shall be no less than the fluid temperature. Leakage and actuator forces shall be measured.

- E. Each component of the complete fluid systems test apparatus shall be identified by a code number which shall be scribed on the component. A complete log shall be maintained on each component to include the following: Manufacturer's designation and specifications; materials certification report; inspector's report; record of all tests (time and conditions); record of all post-test inspection reports, including photographs and failure analysis where applicable; record of all repairs and substitution of new components. These logs shall be updated at weekly intervals and maintained in a file which is available for inspection by the NASA Project Manager.

TASK II - Materials Selection, Procurement and Testing

- A. The following classes of materials shall be considered for seals and/or gland bearing materials. Ten materials shall be selected by the Contractor with the approval of the NASA Project Manager. The selected materials shall be obtained and formed into appropriate test specimens for a one inch rod seal. In all cases, unless specifically approved by the NASA Project Manager polished hard chromium plating shall be used for the mating surfaces.
1. Polyimide high temperature polymer (unfilled and metal filled)
 2. Silver-metal composites developed by Illinois Institute of Technology under Air Force Contract No. AF33(616)-7310
 3. Silver-polymer composite (Polymet)
 4. Silver-base alloys or other soft phase duplex structures
 5. Other metallic matrix materials
 6. High strength metals (steel, titanium, cobalt, etc.)
 7. New types of high temperature elastomeric materials
 8. High temperature carbon graphite
- B. Tests measuring bearing characteristics, hardness, elasticity, mechanical strength and other mechanical properties important to hydraulic system seals shall be made on the selected materials. All properties shall be determined at the projected maximum operating temperature, except hardness. Chemical compatibility tests with five fluids selected by the Contractor and approved by the NASA Project Manager shall be made at temperatures of 400°, 500° and 600°F. It is anticipated that these five fluids will be of the following types:
1. Chlorinated phenyl methyl silicone, General Electric Co. - F50
 2. Super-refined mineral oils, MLO 60-294.

3. Monsanto Co. MCS 293 modified polyphenylether.
4. Monsanto Co. MCS 310 Halogenated polyaryl fluid.
5. DuPont fluid PR-143-AB, fluorocarbon.

Using the results of these tests the Contractor shall select five materials from the ten materials tested for further investigation under the following TASKS. The five materials selected shall be subject to the approval of the NASA Project Manager.

TASK III - Seal Design Development

- A. Seals shall be designed which most effectively use the mechanical properties of the individual materials selected in TASK II for further investigation. Design studies shall provide for such consideration of the selected materials as to give optimum rod-end seal designs for each material. Such designs may logically provide for spring mounting to compensate for reduced elasticity, pressure balancing to improve endurance and the use of coatings or films as needed because of varied conformability. The seal designs shall be subject to the approval of the NASA Project Manager.

TASK IV - Low Pressure Tests

- A. The five seal materials and designs shall be tested in the one-inch diameter 100 psi pressure rod and seal test facility described in TASK I. The best seal material and designs from one (1) inch diameter test shall be evaluated in three (3) inch diameter seals under otherwise identical conditions. The test fluid shall be chlorinated phenyl methyl silicone unless the NASA Project Manager directs that another fluid shall be used instead. Operation shall be for 50 hours (or until seal failure, if less than 50 hours) at each of the fluid temperature levels 400°, 500°, and 600°F. Operation shall alternate between operation at 30-40 CPM with ± 2 to ± 4 inch stroke and operation at 100-300 CPM with $\pm .05$ to $\pm .10$ inch stroke. Seal leakage and actuator forces shall be measured. Seal leakage in excess of one drop per minute or a two-fold increase in required operating force shall be criteria for seal failure. These criteria may be modified with the approval of the NASA Project Manager.

TASK V - High Pressure Tests

- A. Three materials selected by the Contractor from the results of the low pressure tests and approved by the NASA Project Manager shall be tested in the 0 to 4000 psi rod seal test unit described in TASK I. These tests shall run for a total of 3000 hours or until

seal failure occurs. A single test apparatus capable of both the low pressure tests described in TASK IV and the high pressure tests to be described in this TASK V may be used. Rod end seals of 1 inch diameter and rod end seals of 3 inch diameter shall be tested concurrently in the same unit at a pressure of 3000 to 4000 psi with the following operational cycle:

1. Operation for 35 minutes at 15-20 CPM alternately using $\pm 1/2$ to ± 1 inch stroke and ± 2 to ± 4 inch stroke. Fluid Temperature shall increase approximately linearly from 100°F to 500°F during this period. Ambient temperature shall be increased in a 150 minute period.
 2. Operation for 125 minutes at 100-300 CPM using $\pm .05$ to $\pm .10$ inch stroke. Fluid temperature shall be at 500°F.
 3. Operation for 20 minutes at 15-20 CPM alternately using $\pm 1/2$ to ± 1 inch stroke and ± 2 to ± 4 inch stroke. Fluid temperature shall decrease approximately linearly from 500°F to 100°F during this period.
- B. In addition, a single-stage high pressure one (1) inch diameter seal, fabricated from the best material under Para. "A", TASK V, shall be developed and evaluated in the endurance rig described in TASK I, at:
1. Temperature, 500°F.
 2. Pressure, 3000 psi, maximum.
 3. Test profile described in TASK V, Para. A.
 4. The seal shall be designed and tested to failure or 100 hours with subsequent redesign and testing subject to approval of the NASA Project Manager.
- C. For all operations seal temperatures shall be no less than the fluid temperatures. Following every 20 such cycles the seal assembly shall be subjected to a 4 hour cold-soak at -40°F ambient. The leakage in the cold system shall be checked at the end of the soak period and continuously during warm-up prior to the subsequent operational cycle.
- D. The operational cycle described above shall be considered typical of test requirements but shall be subject to redirection by the NASA Project Manager. The test fluid shall be chlorinated phenyl methyl silicone unless the NASA Project Manager directs that another fluid shall be used instead.
- E. Seal failure criteria sufficient for the termination of a run are leakage in excess of one drop per minute or a two-fold increase in operating force.

APPENDIX C SOURCE OF MATERIALS

Polymer SP	E. I. DuPont de Nemours and Co. Inc.
Polymet	The Polymer Corporation
Silver alloy	Handy and Harmon Co.
Nickel Foametal	Metallurgical Department General Electric Company
Westinghouse Composite	Westinghouse Electric Corp.
Metco flame-plate	Metco Company
Vascojet 1000	Vanadium - Alloys Steel Company
Silver impregnated fiber composite	Republic Aviation Division Fairchild Hiller
Cobalt-Molybdenum alloy (75% Cobalt, 25% Mo)	NASA-Lewis Research Center
G. E. F-50 silicone	General Electric Company
MCS-293 MCS-3101	Monsanto Chemical Co.
PR-143AB	E. I. DuPont de Nemours and Co. Inc.
MLO-60-294	Humble Oil & Refining Co.
Graphitar	United States Graphite Co.